Book II

State of California
THE RESOURCES AGENCY
Department of Water Resources

BULLETIN No. 164

TEHACHAPI CROSSING DESIGN STUDIES

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MAY 1965

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TEHACHAPI PUMPING PLANT

COMPARATIVE ANALYSIS OF LIFT CONCEPTS

PUMPS AND INTERFACE ELEMENTS

VOLUME II

TECHNICAL STUDIES

April 1965

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Engineering Division
Los Angeles

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CHAPTER 1

MODEL TESTING PROGRAM

A. PURPOSE AND SCOPE

The pumps to be used in the pumping station for the Tehachapi Crossing for the California Aqueduct will represent an important achievement in the field of pump engineering, because of the combined requirements of high head and high flow. A preliminary study of existing pumping machinery in this country and in Europe reveals that there are several types of centrifugal pumps able to perform this task, depending on lift concept. Each lift concept proposed calls for a different pump design, this along with the size of each unit stresses the importance of testing a model of each pump to determine the most feasible design. Model testing provides the most economical way of investigating and evaluating the proposed pumps from the standpoint of performance and operational requirements. Actual tests shall be conducted on each model pump to ascertain the following:

1. Head, capacity, efficiency, horsepower, NPSH
2. Normal and abnormal operating conditions (three-quadrant testing)
3. Pressure fluctuations
4. Inlet velocity distribution
5. Casing and diffuser loads
6. Pump and shaft vibrations
7. Thrust unbalance.

B. SELECTION OF MODEL TESTING FIRMS

The model pumps for each lift concept are to be tested by the firms contracted to perform the design and fabrication. The selection of these firms was made by rating each firm against the other with respect to seven items of criteria:

1. Time

2. Management plan

3. Program approach

4. Manufacturing facilities

5. Test facilities

6. Pumps and turbines constructed

7. Cost

Selection was made from 13 pump manufacturers and two independent testing laboratories. A complete report describing the evaluation procedure, method of rating each firm and the results of these ratings can be found in Daniel, Mann, Johnson, & Mendenhall's report entitled "Tehachapi Pumping Plant Research & Development Program Evaluation of Model Testing Firms", dated 2 March 1964.

Based on the results of this selection analysis, DMJM has entered into contract with three model pump firms: 1) Allis-Chalmers Mfg. Co./Sulzer Bros., 2) Baldwin-Lima-Hamilton/J. M. Voith, G.M.B.H., and 3) Byron Jackson Division, Borg-Warner Corp.

C. CONTRACTS AND SPECIFICATIONS FOR TESTING

Contained within each of the contracts written by DMJM to the model pump firms are three sections governing the manner in which the model pumps are to be tested. Section 4, Test Facilities and Equipment, describes the minimum requirements concerning test equipment and test instrumentation. Section 5, Detailed Test Procedure, specifies the tests to be conducted and references the applicable test codes. Section 6, Processing and Data Format, itemizes the essential factors in order to standardize the presentation of data. These sections are identical for each of the model pump firms with one exception; Section 5, Paragraph 5.01 of the contract to Byron Jackson specifies:

"These tests shall be made at a head equal to the prototype head." The other two firms are to conduct tests at "a head not less than 65 per cent of the prototype head." Exhibited here is a copy of the abovementioned sections which act as the controlling documents for conducting the model tests since they are a part of the contract:

(Excerpt from model test firm contract)

SECTION 4

TEST FACILITIES AND EQUIPMENT

4.01 GENERAL

The Contractor shall furnish all necessary facilities, equipment, instruments and other required items for performing the tests specified in Section 5, "Test Procedures". Unless otherwise specified, the equipment and instruments shall conform to the applicable requirements of ASME Power Test Code for Centrifugal Pumps PTC 8.1-1954 or Standards of the Hydraulic Institute, Centrifugal Pump Section, 1961 Revision.

4.02 PRECISION AND ACCURACY

The test facilities and equipment proposed for use by the Contractor will be acceptable provided they meet all requirements of precision and accuracy to obtain highly reliable model performance. The statement of precision and accuracy of the various items of equipment, including all measuring instruments, submitted by the Contractor with his proposal shall be fully supported and guaranteed. In no case, shall the precision and accuracy be less than the requirements of the referenced ASME Code and Hydraulic Institute Standards. In general, it is expected that minimum precision and accuracy of instruments will be such as to insure an overall efficiency f_p of ± 0.3 per cent for zone of normal pump operation. The inaccuracy, f_p, of the pump efficiency determination is computed from the individual inaccuracies as follows:

$$f_p = \pm \sqrt{f_Q^2 + f_H^2 + f_T^2 + f_n^2}$$

Where the subscript Q refers to Capacity

Where the subscript H refers to Head

Where the subscript T refers to Torque

Where the subscript n refers to speed

4.03 APPROVED FACILITIES AND EQUIPMENT

The Contractor, prior to commencement of tests shall submit for approval a final corrected listing of the facilities, equipment, instruments and other items required for the different tests, giving the manufacturer's name and model, performance data, and other information including photographs, sketches, and lists.

(Excerpt from model test firm contract)

A drawing of the complete test setup for each of the different kinds of tests required — performance, efficiency, cavitation, velocity and pressure distribution shall be included in the data.

4.04 CONDITION OF WATER

The water shall be clean and clear and free of solid ingredients. Water source and quality shall be subject to the approval of the Engineer. Free gas particles or air bubbles shall be removed to the extent possible. The air and gas content and the temperature shall be measured frequently. During each test run, the change in gas content and in temperature shall be at a minimum. The temperature of the water shall not exceed 85°F.

4.05 FLUCTUATIONS DURING TEST

During test, the fluctuations shall not exceed those listed in the ASME Test Code PTC 8.1, Section 3-48.

4.06 MODEL INLET

The flow at the inlet to the model shall be free from vortices. A model of the suction piece ahead of the pump shall be included in the test setup.

4.07 MEASUREMENT OF WATER DISCHARGE

The primary water measurement shall be by Venturi meter. Calibration of the flow measurement equipment shall be accomplished in situ by weighing tank or volumetric tank and under prevailing test conditions unless otherwise approved by the Engineer. It is desirable to have a secondary measuring method in order to check the satisfactory operation of the flow measuring equipment.

4.08 PRESSURE MEASURING INSTRUMENTS

Primary discharge pressure measurement shall be by dead-weight tester. Primary inlet pressure measurement shall be by either dead weight or liquid manometer. It is desirable to have secondary measuring instruments in order to check the primary measurements for the same quantity. Only the primary measurement shall be considered valid. The selected instrument shall be clearly designated in the program.

4.09 WATER LOSSES

Where the water loss through the shaft gland is appreciable, it shall be taken into consideration in efficiency determination. Stuffing boxes without physical contact will be preferred for the model.

(Excerpt from model test firm contract)

4.10 TORQUE

Torque measurements shall include separate measurements of bearing friction for running and breakaway conditions.

4.11 SPEED

Rotative speed shall be measured by electrical counter. Additionally, a direct reading instrument shall be used to check on the constancy of the speed.

(Excerpt from model test firm contract)

SECTION 5

TEST PROCEDURES

5.01 DETAILED TEST PROCEDURE

The Contractor shall perform the following model tests in accordance with the test procedures specified herein. Pump performance tests shall be made under normal operating conditions to obtain the H-Q curbes from cut-off to point of significant cavitation, and the power input, efficiency and NPSH curves superimposed on the H-Q plot. These tests shall be made at a head not less than 65 per cent of the prototype head. Specifically, the tests shall include:

a. Preliminary tests

b. H-Q efficiency characteristic tests

c. Discussion of test results

d. Improvement of efficiency characteristic, if necessary, by impeller and diffuser modifications and re-test.

e. Cavitation tests on suction impeller under approximately full prototype head per stage.

f. Improvement in cavitation performance if necessary and re-test.

g. Discussion of test results

h. Report

i. Make complete pump performance tests under normal and abnormal conditions to obtain a complete three-quadrant characteristic plot including the following operational conditions:

1. Normal pump (positive head, positive rotation, positive flow and positive torque).

2. Normal rotation pump with reverse flow (positive head, positive rotation and positive torque; negative flow).

3. Normal turbine (positive head, positive torque, negative rotation and negative flow).

(Excerpt from model test firm contract)

j. Make tests to determine pressure fluctuations in the casing waterway passages.

k. The velocity distribution and the pre-rotation of the flow ahead of the impeller inlet when pumping at operating head shall be determined.

l. Stress analysis shall be made by analytical calculations to determine the stresses that may be expected in the prototype at critical places. Stresses in the casing and blade loadings of the diffuser shall be investigated.

m. Vibration Analysis. This analysis includes the identification of all significant modes of vibration in the prototype, including the vibration due to passage of the impeller vanes past diffuser vanes, the vibration due to shaft deformation, and any other mode to which the particular pump design may be sensitive.

n. Thrust Unbalance. The model shall be tested to determine the axial and radial thrust unbalance during normal operation using approved procedure.

5.02 SUBSIDIARY SPECIFICATIONS

a. General

Unless otherwise specified in detail or approved, the Contractor shall perform the tests in accordance with the applicable procedures given in ASME Power Test Code for centrifugal pumps PTC 8.1-54 and the Standards of the Hydraulic Institute, Centrifugal Pump Section, revision dated January, 1961. In lieu of meeting these requirements, approval may be given to the test procedures submitted by the Contractor provided that such procedures will, in the opinion of the Engineer, give results of an equal or higher degree of accuracy.

b. Calibrations

All measuring equipment shall be calibrated carefully before and after tests, and a record of such calibrations, with the number of times required during test operations, shall be kept and included in the final report submitted at the end of the work by the Contractor. Calibration shall be performed in accordance with the procedure described in the "Information to be Provided by the Proposing Firm," submitted with the proposal. The probable error of measuring equipment submitted by the Contractor in connection with the furnishing of test equipment covered hereinabove shall take into account any drift from the null point after calibration.

(Excerpt from model test firm contract)

c. Operating Tests

The tests to obtain a complete three quadrant characteristic in the zone of energy dissipation and turbine operation shall be made similar to tests made by R. T. Knapp at California Institute of Technology and described in ASME Transactions dated November, 1937. Commercial accuracy only will be required for these tests. The tests shall be carried a step further to separate the total pump losses into hydraulic, volumetric and mechanical losses for the model and such expected losses for the prototype.

d. Cavitation Tests

These tests shall conform to the third arrangement shown in ASME Power Test Code for Centrifugal Pumps PTC 8.1-54, Article 89c. (This arrangement permits the test to be made with only the suction head varied and the pump head maintained constant.) However, if the Contractor wishes to use any other procedure, he shall submit evidence to the Engineer that the results obtained are equally satisfactory for the purpose, and receive approval to use such substitute procedure.

A transparent window must be provided in such manner and location that the inlet to the first stage impeller may be observed to permit visual check for evidences of cavitation. Necessary photographs shall be taken and shall be included in the report.

e. Test to Determine Pressure Variations

These tests shall be similar to, and no less precise than the tests described in ASME Paper No. 63-AHGT-11. This includes an investigation of the crossovers as well as the volute section.

f. Stress Analysis

Stress analyses shall be made to determine the stresses that may be expected in the prototype at critical places, including stresses at over-speed conditions. Stresses in the casings and diffusers shall be investigated and reported. No stress testing of the pump model shall be required.

(Excerpt from model test firm contract)

SECTION 6

PROCESSING AND FORMAT OF DATA

6.01 GENERAL

Unless otherwise specified or approved, the Contractor shall conform to the provisions of the ASME Power Test Code for Centrifugal Pump PTC 8.1-1954, or those of the Standards of Hydraulic Institute, Centrifugal Pumps section, November, 1961 Revision, applicable for the processing and format of data. This includes the procedures for correcting data, and computing pump performance values from direct measurement data, the "affinity" rules for obtaining prototype performance values from model performance, and the plotting of performance.

6.02 CORRECTIONS OF TEST MEASUREMENTS

The following test measurements shall be corrected before further use in determining the essential parameters.

a. Head

The pressure difference produced by the pump between suction and discharge flanges shall be converted into a head expressed in feet of the fluid pumped in order to make the results independent of the temperature and of other influences affecting the density of the fluid.

b. Capacity

The pressure difference at the meter for measurement of capacity in cubic feet per second shall be converted to height of a column of the fluid which is flowing through the meter.

c. Torque

The torque measured on the dynamometer or other approved device shall be converted into that torque which would exist if the water in the pump had a temperature of 68°F. (standard temperature). The reduction of the power to a standard temperature of 68°F. is necessary in order to make power curves obtained under different temperature conditions comparable with each other.

(Excerpt from model test firm contract)

6.03 FORMAT OF DATA

The Contractor shall provide H-Q plots on which efficiency, power and NPSH curves are added, that represent the predicted performance of the prototypes. These plots shall be accompanied by an explanatory sheet that notes all the factors used in developing the plots from the direct measurements of data on the models.

The Contractor shall provide a complete three-quadrant characteristic plot of the expected values for the prototype.

6.04 GRAPHICAL PRESENTATION

All curves shall be plotted on 8-1/2" x 11" graph paper or larger, prepared in the English language and using the English system of measurement. See also Section 8.

D. MODEL PUMP DESIGNS

The design of each of the three model pumps is quite different since each model pump firm has proposed a pump suitable for a different lift concept. The model pump designed by Allis Chalmers/Sulzer for the single-lift concept is a four-stage, single suction centrifugal pump. Baldwin-Lima-Hamilton/Voith model pump for the two-lift concept is a two-stage, double suction centrifugal pump Both of these contractors have also built a single-stage, single suction model to check the cavitation behavior of the impellers For the three-lift concept, Byron Jackson has designed a single-stage, single suction centrifugal model pump. Each of the model pumps will be tested in the horizontal position, mainly because of convenience in test loop arrangement.

1. Single-Lift Concept - A-C/Sulzer Four-stage Pump

Allis Chalmers/Sulzer has submitted the following description and drawings of their model pump:

Material prepared for this spot falls within the category of information considered proprietary by Daniel, Mann, Johnson, and Mendenhall and has been deleted at the request of that firm.

E. TEST FACILITIES AND PROCEDURES

Data furnished by each model test firm is presented in Volume I, Chapters 6, 7, and 8, and in the following sections:

In order to insure that the results from each laboratory are comparable with the others to the degree that is practical, the ASME Power Test Code and the Hydraulic Institute Code have been referenced in the model contracts and certain specific instrumentation and test methods have been specified.

Each of the test firms has submitted equipment lists and descriptions along with test procedures to be employed for the various kinds of tests.

As a part of his duties as a Consultant to DMJM, Professor L. J. Hooper has reviewed the procedures and equipment lists and has personally visited each of the testing laboratories in order to ascertain compliance with contract requirements and determine that satisfactory tests will be obtained.

The lists and procedures are presented in the following section:

1. A-C/Sulzer - Four-stage Pump Model

a. Test Equipment

The following is a list and description of the test equipment and instrumentation to be employed, while conducting the model performance tests:

"TEST EQUIPMENT DATA SHEET

TESTING FIRM	SULZER BROS. LTD.
Model Pump Driver	
Type and Nameplate	Dynamometer motor
Data of Dynamometer motor	Three-phase asynchronous motor, slip-ring type - Brown Boveri & Co. Type MDS: 242 SOSD

Output (KW):	1620 kW continuous rating)) at 2950 rpm 1840 kW for one hour)
Speed (RPM):	2950 rpm
Possible Speed Variation	2950 rpm to 2200 rpm 10 rpm liquid resistance
Torque Measurement	
Method & Instrument type:	Dynamometer (by weights)
Range:	Zero to 4400 foot pounds
Accuracy:	± 0.1%
Method of Calibration	Torque measured directed by calibrated weights. The bearing losses will be excluded by running of the shaft without impellers and measuring the torque. This part of the calibration is done before and after all testing.
Speed Measurement	
Method Instrument type:	Photo-electric cell and electronic counter. Cell: Type B-752 No. 1 counter: Type B-731 No. 1 Manufactured by: Ebauches S. A. Meuchatel Switzerland
Range:	Zero to 99,000 rpm

Accuracy:	± 0.1%
Method of Calibration:	Accuracy of time base controlled by a Quartz clock.
Secondary Measurement:	Hand tachometer, type Hasler, Bern.
Flow Capabilities	
Maximum available flow as limited by:	
Booster Pump	23 cfs
Type of pump:	SP - 40 - 45, Sulzer Bros. (Mixed flow: Helimax; single-flow, one-stage; n = 750 rpm)
Flow Measurement:	
Instruments:	a) ASME-nozzle, high - β - series β = 0.7. Pressure taps: Inlet connections - 1 pipe diam. (D) preceding the entrance plane of the elliptic section of the nozzle, (pipe wall taps). Outlet connections 1/2 pipe diam. following the entrance plane of the elliptic section of the nozzle (throat taps). See ASME: Power test code, 19.5; 4-1959, Section 3, subsection B, Par. 25 - 30. b) VDJ-Diaphragm, β = 0.58, corner taps See: VDJ-durchfluss-Messregeln DIN 1952 Aug. 1943. Readings for nozzle and diaphragm: Mercury or water differential manometers
Accuracy:	± 0.25%

Range:	a) 4 c.f.s. ÷ 12 c.f.s. up to 21 c.f.s. 1 ft. of water ÷ 6.5 ft. of water ÷ 1.6 ft. of mercury. b) 1.0 c.f.s. ÷ 6.0 c.f.s. 1.0 ft. of water ÷ 6.5 ft. of water
Calibration:	By volume in a tank of 43,000 gallons. The arrangement for calibration is shown in Drawing 400.9.335.015 { FIG. No. 6-6 }. A booster pump transfers water from the main reservoir into a weir tank. A round weir in this tank provides a constant suction head for the main pump. After passing the pipe system, which includes the nozzle or the diaphragm, the flow can be led either to the main reservoir or to the calibration tank. The minimum time required to fill the calibration tank will be 100 seconds. The nozzle differential pressure will be constantly read during the measurement period.
Head Capabilities	
Maximum permissible Pressure in Discharge Line:	910 psi
Discharge Pressure Adjustable by:	Throttling valve (tapered gate valve)
Discharge Pressure Measurement	
Instrument:	Dead weight gauge tester (Sulzer/Haenni-system)
Range:	0.085 ft. of water - 3000 ft. of water
Accuracy:	± 0.2 psi

Calibration:	Control of piston dimensions	
Secondary Measurement:	Bourdon-type gauge	
Max. & Min. Suction Pressure and Relationship to Flow:	52 ft. 0.5 cfs	43 ft. 21 cfs
Suction Pressure Adjustable by:	Throttling valve (tapered gate valve)	
Positive Suction Pressure applied by:	Booster pump	
Suction Pressure Measurement:		
Method and Instrument:	Dead weight gauge tester or mercury, column, as required.	
Accuracy	$\pm$ 0.2 psi	$\pm$ 0.1 psi
Secondary Measurement:	Mercury column	
Accuracy of total head measurement	$\pm$ 0.1%	
Temperature - Measurement of air, water, oil		
Instrument:	Mercury thermometer	
Leakage losses:	Calibrated tank, volume 130 gallons	

"

A plan view drawing of the Sulzer test facility is shown in FIG. 1-11, Sulzer Drawing 400. 9. 335. 013. Sectional views of this facility are shown in FIG. 1-12, Sulzer Drawing 400. 9. 335. 014. FIG. 1-13, Sulzer Drawing 400. 9. 335. 015, is a diagram showing the piping arrangement of the test bed

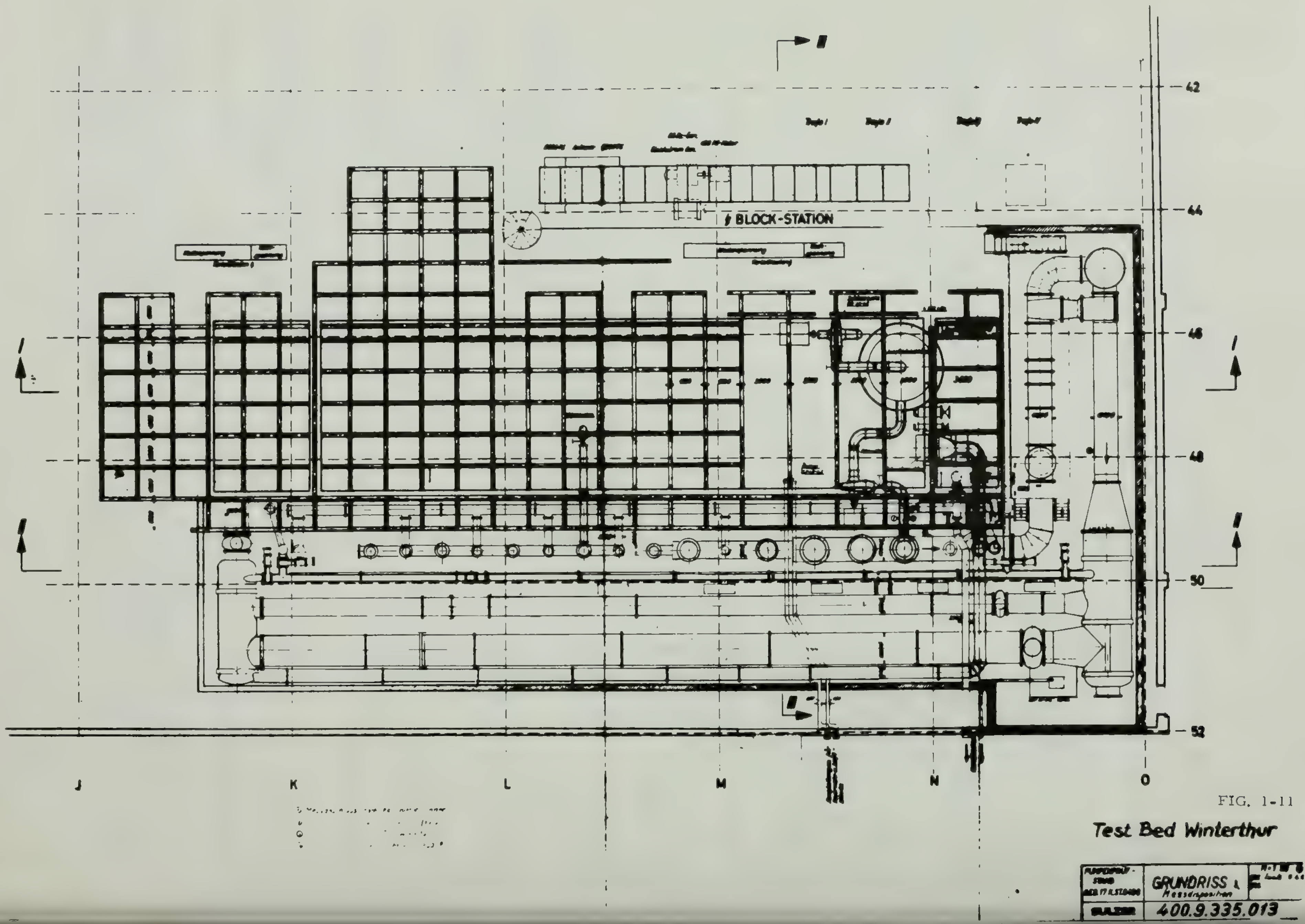


FIG. 1-11

Test Bed Winterthur

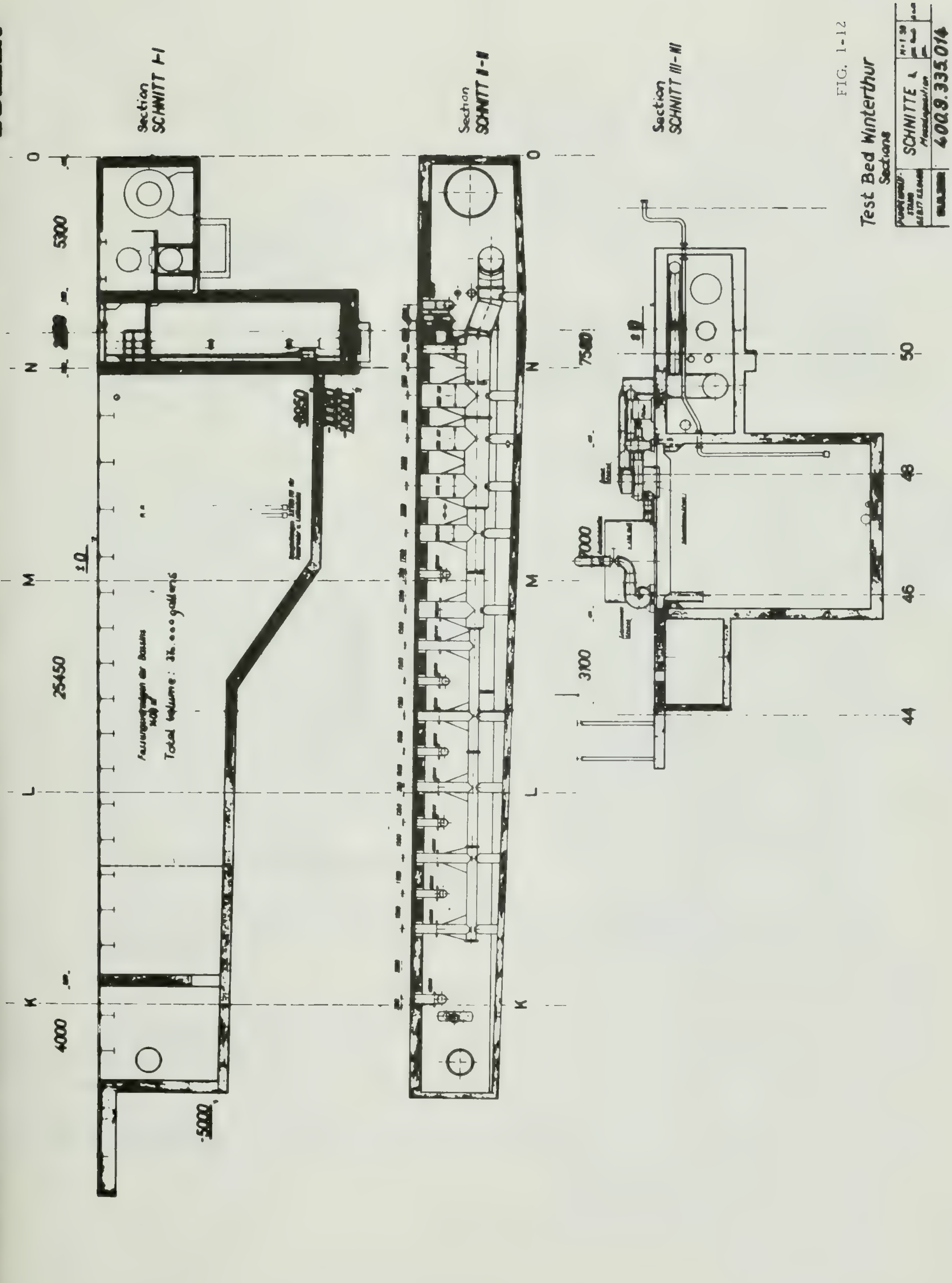
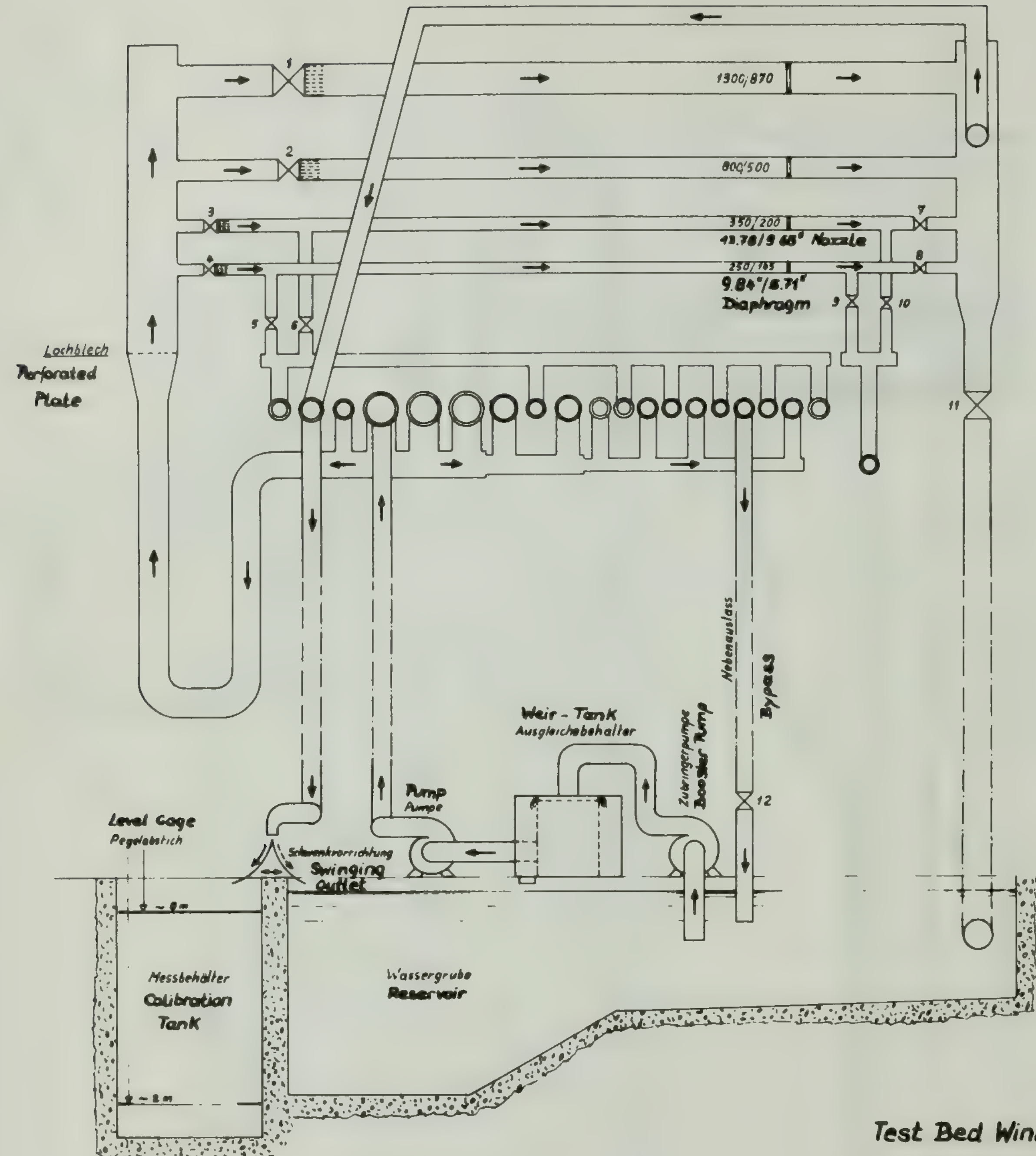

SULZER
Section
SCHNITT I-I
Section
SCHNITT II-II
Section
SCHNITT III-III
5300
25450
4000
7500
3100
Total volume
Test Bed Winterthur
Sections
SCHNITTE
4009.335.014

FIG. 1-12



Test Bed Winterthur
Sulzer

Pumpenprüfstand W.thur
Hydraulisches Schaltschema
für Blendenmessung

400.9.335.015

FIG. 1-13

and basement depicting the flow path and method of flow measurement. The model pump test set-up is shown with piping dimensions in Chapter 6 of Volume I. The booster pump transfers water from the main reservoir to the inlet of the model pump through a pipe of 15.75 inch diameter, containing a gate valve, a reducing cone, a 7.7 foot length of 11.8 inch diameter pipe, a 1.64 ft. length of vaned straightener and then 4.2 feet of 11.8 inch pipe before the inlet bend.

The four pressure taps for suction head measurement are situated 2.95 ft. behind the straightener and 1.31 feet before the inlet flange of the model pump. The four inlet pressure taps are connected to the manometer through a manifold loop. It is possible to check each pressure tap alone, while the other three are closed by valves.

The model pump and the dynamometer motor are rigidly fastened on steel rails. The outlet flange of the model pump is connected with the water-measuring system by pipes of several diameters. The first section of pipe is a straight length, 9.85 inches in diameter, and contains the outlet pressure taps at a distance of 3.3 feet from the outlet flange of the model. A cone leads to a pipe of 5.9 inch diameter. The throttling valve is installed 7.22 feet downstream of the pressure taps and is followed by an energy absorber. The connection from this point to the inlet flange of the underground-pipe-system is accomplished by a pipe of 11.8 inch diameter. The flow will be directed to either of two meter runs; (1) the 9.8 inch diameter pipe containing the VDI diaphragm for the lower flow rates or (2) the 13.8 inch diameter pipe containing the ASME nozzle for the higher flow rates. The reading of the pressure difference will be done by either water or mercury columns. The flow of water is led back to the main reservoir beneath the surface level, so that a minimum of air is entrained.

b. Model Pump Test Procedures

Allis-Chalmers/Sulzer have submitted test procedures describing the following required model pump tests:

(1) Model pump efficiency test

(2) Cavitation test

(3) Radial thrust test

These test procedures are exhibited as follows:

"REPORT OF PUMP MODEL EFFICIENCY TEST PROCEDURE

The calibration of the nozzle, dead weight manometers and the bearing losses are accomplished before the start of testing. Just before the beginning of a test the following work has to be done:

a. Checking zero-points of instruments including dynamometer motor checking.

b. Measuring elevation of instruments, if necessary.

c. Reading of barometric pressure.

d. Starting booster pump.

e. Starting model pump.

f. Filling the lines between pressure taps and manometers with liquid.

g. Operation of model pump at the chosen point of the HQ-characteristic for a sufficient time so that all values become constant.

h. Balancing the dynamometer motor approximately.

i. Reading of water - and air-temperature.

j. Taking a water-sample (as required) for the determination of air-content.

After the above activities are complete, the final measurements can be taken. These activities are:

a. Exact balancing of the dynamometer motor and reading.

b. Reading of the electric counter for the number of revolutions; check by hand-tachometer.

c. Reading of the differential pressure given by the nozzle.

d. Reading of the dead weight gauge tester and Bourdon gauge, connected with the outlet-line of the pump.

e. Reading of the dead weight gauge tester and/or the mercury column connected with the suction line of the pump.

f. Reading of water-temperature.

g. Reading of oil-temperature.

h. Reading of oil-capacity.

i. Measuring the labyrinth-leakage losses by taking the time for a certain level difference in a calibrated tank.

All readings will be taken in metric units. The necessary corrections will be made in metric units and the changes to English units are only executed with the final results of H and Q and brake horsepower. All readings will be taken at least three times for one test point during a time of about two minutes. As far as possible, the readings of the values are taken simultaneously. The test personnel will normally be two to three men who are familiar with test-bed-work and handling of the instrumentation. A first evaluation of the required values is made just after the test while the pump is set on another point and is coming to constant conditions. It is planned to measure 12 points or more for a complete HQE-performance curve, so that head and efficiency are known over the entire pump range.

Additional points are chosen close to the best efficiency point so that it can be well defined. After finishing a complete HQE-characteristic, the following work has to be done:

a. Stopping model pump and booster pump.

b. Reading of air-temperature.

c. Taking a water-sample for determination of air-content (as required).

d. Checking zero-points of instruments including dynamometer motor checking.

Precise calculations are made after completion of testing, using more accurate calculating methods:

"REPORT ON CAVITATION TEST PROCEDURE

Purpose of Tests

The purpose of this series of tests is to determine the cavitation coefficients for the model pump. Two tests will be performed. They are:

a. A single stage pump running at 2950 RPM. The performance will be:

13.10 cfs

488 ft. head

615 kw

See enclosed drawings:

400.0.110.944 - Single Stage Model Cross section

400.0.110.047 - Single Stage Model General Arrangement

400.9.335.027 - Single Stage Model Test Arrangement

{These drawings are presented in Chapter 6 of Volume I}

b. The four-stage model running at 2390 rpm. The performance will be:

10.56 cfs

1287 ft. head

1308 kw

See enclosed drawing:

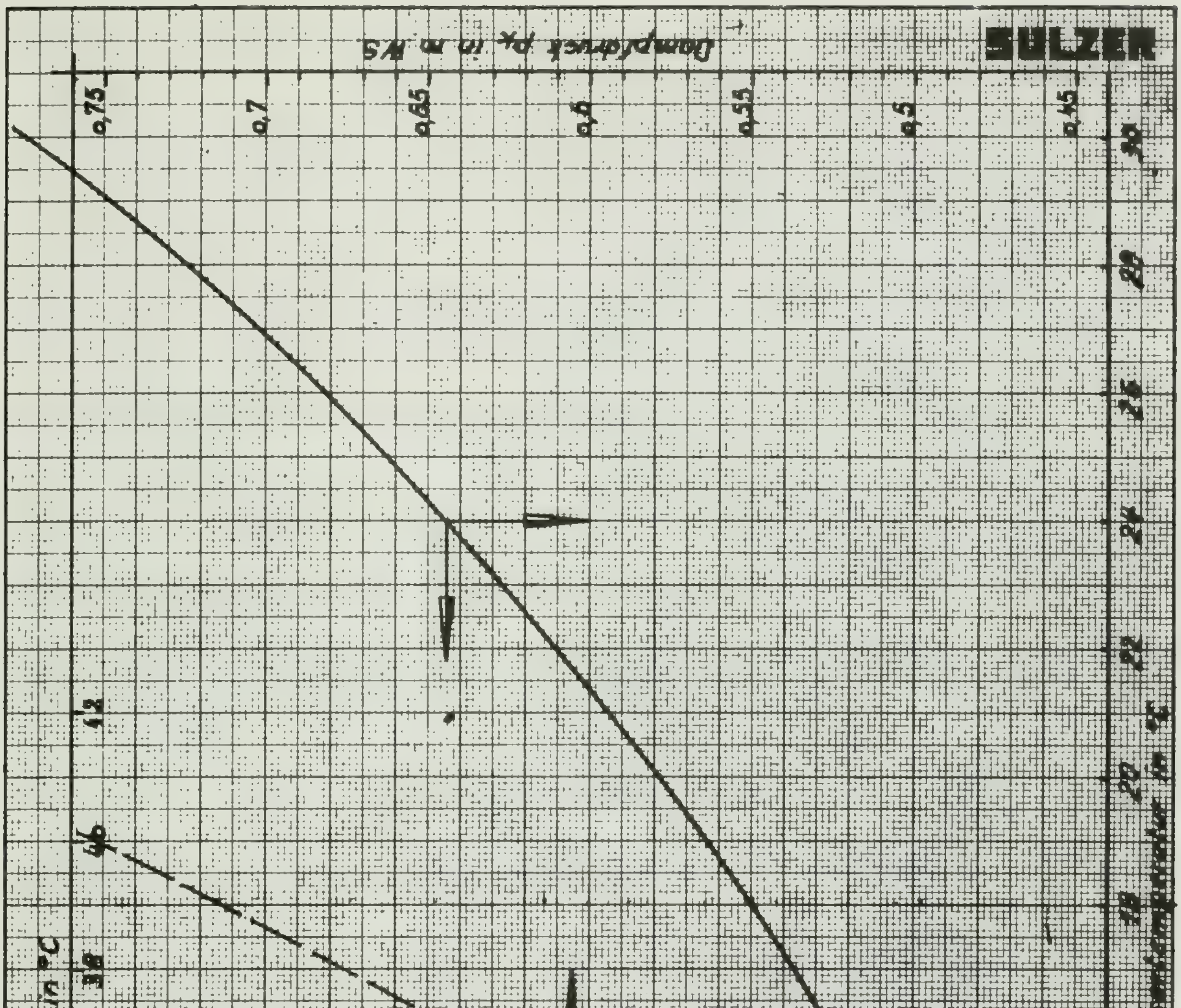
400.0.110.046 - Four-stage Model General Arrangement {Vol. I, Ch. 6}

Test Quantities & Nomenclature & Notation

H_b — Barometric pressure in meters of water measured at air temperature.

H_t — Vapor pressure in meters of water measured at temperature of water.

See enclosed curve PE-4877. {FIG. 1-14}



Dampfdruck p_k des Wassers.

FIG. 1-14

Anlage:	Best.Nr.	Datum: 25.11.49	Name:	PE 4877

$S_2 = \frac{\pi}{4} D_L \quad = 0.049\,m^2$ — Area of discharge measuring cross section.

$D_{L_2} = 0.25$ — Discharge measuring cross-section diameter-meters.

$P = KGN$ — Pump power input as measured by dynamometer motor — KW.

K — Dynamometer motor constant.

G — Weight on dynamometer motor arm — Kg

N — Pump RPM as measured by electronic cell and counter.

$\eta = \dfrac{QH}{kP}$ — Pump efficiency.

k — Calculation constant.

Test Procedure

The cavitation tests will be carried out at the following rates of flow:

a. Single stage model at 290 RPM

Flow — cfs	10.5	11.8	13.1	14.4	15.7
Percent of Rated	80	90	100	110	120

b. Four stage model at 2390 RPM

Flow — cfs	8.45	9.5	10.56	11.6	12.7
Percent of Rated	80	90	100	110	120

The pump suction pressure will be varied by throttling the valve in the suction piping (see 400. 9. 335. 027). {Vol.I, Ch.6}. The aforementioned readings will be taken and the NPSH calculated as follows:

$$NPSH = H_B - H_T \pm H_1 + \frac{C_1^2}{2g} \text{ (meters)}$$

T_w	— Water temperature at pump inlet - °C.
$-H_1$	— Static pressure at pump inlet in meters of water. Measured by use of Hg manometer and at 4 points in measuring cross section.
$+H_2$	— Positive suction pressure at pump inlet in meters of water. Measured by use of Haenni/Sulzer dead weight gauge tester and at 4 points in measuring cross section.
$\frac{G_{air}}{G_{water}}$	— Air content of water - percent.
$C = \frac{Q}{S_1}$	— Velocity in inlet measuring cross section - m/s.
$\frac{C_1^2}{2g}$	— Velocity head in meters of water at pump inlet measuring cross section.
Q	— Pump flow in $\frac{m^3}{sec}$ measured with calibrated nozzle.
S_1	— $\frac{\pi}{4} D_{L_1}^2 = 0.096\, m^2$ – Area of inlet measuring cross section.

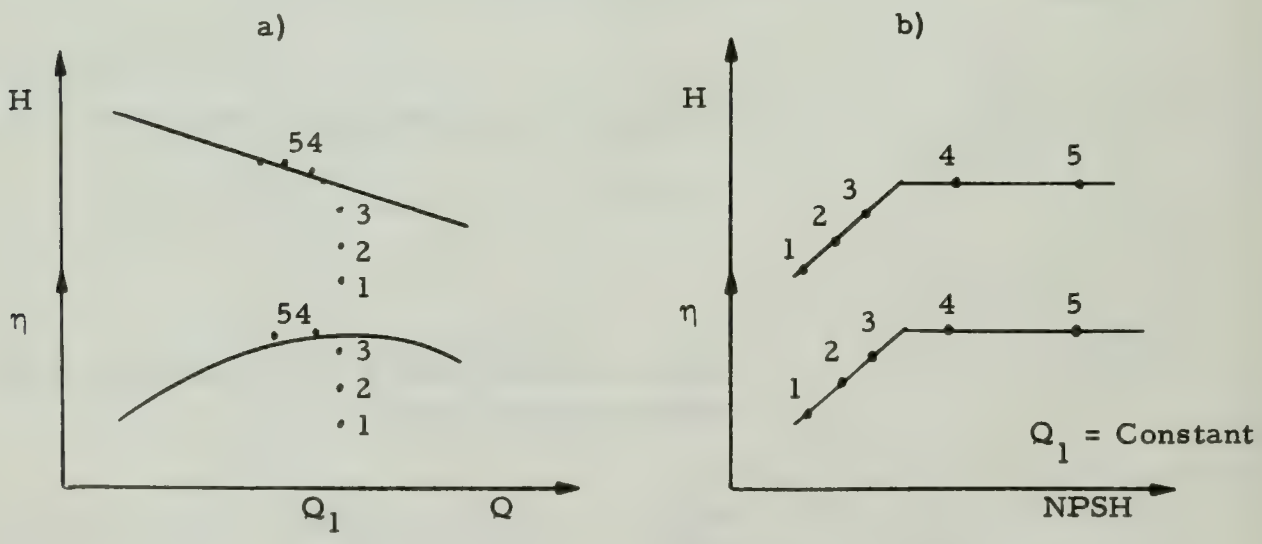
$D_{L_1} = 0.35$ — Inlet measuring cross-section diameter-meters.

H — Static pressure in meters of water at pump outlet. Measured by use of dead weight gauge tester and at 4 points in measuring cross section.

$C_2 = \dfrac{Q}{S_2}$ — Velocity in $m/_s$ in discharge measuring cross section.

$\dfrac{C_2^2}{2g}$ — Velocity head in meters of water in discharge measuring cross section.

The calculated values of H, NPSH and η will be plotted as illustrated below:



Sketch 'a' shows the typical characteristic curve with 'break-off' curves superimposed. Sketch 'b' is a plot of H and η against NPSH. The value of critical NPSH is the point where H and η decrease from their normal value with a decrease in NPSH.

In addition to this classical method of determining the NPSH characteristics of the pump, visual and photographic observations will be made of the impeller inlet at many flow rates and NPSH values. A strobolight will be placed against one observation window to illuminate and 'stop' the motion of the inlet of the impeller and flow and photographs will be taken through a second window. See Drawing 400.0.110.044. {Vol. I, Ch. 6}

The NPSH values as obtained from sketch 'b' will be plotted as a cavitation coefficient K^1 vs. capacity coefficient ϕ. These are defined as follows:

$$K = \frac{NPSH}{V_2^2/2g}$$ — Cavitation coefficient

$$\phi = \frac{Q}{S_2^1 V_2}$$

where

$$V_2 = \frac{\pi N D_2}{60}$$ — Peripheral speed at impeller outlet — m/sec.

$$S_2^1 = \pi D_2 B_2$$ — Outlet area of impeller — m^2

D_2 — Outlet diameter of impeller — m

B_2 — Outlet width of impeller — m

Conclusions

The cavitation test results on both the single and four stage model pumps will be plotted as dimensionless coefficients K^1 vs. ϕ and photographs will be taken of the impeller inlet during operation. These data can be used to determine required submergence for the prototype pump.

"REPORT OF RADIAL THRUST TEST PROCEDURE

The radial thrust on the impellers will be measured by determining the circumferential static pressure distribution at the O.D. of the second stage impeller.

Twelve pressure taps (see 400.9.110.378) {FIG. 1-15} have been drilled in the diffuser and will be connected to a dead weight gauge tester. Each tap can be individually connected to the tester by a system of shut-off valves and piping.

The test procedure will be:

a. Bleed all lines between taps and tester.

b. Read barometer, air and water temperature.

c. Read static pressure at each tap.

d. Maintain constant check on pump flow and head to assure constant operation.

A minimum of twelve readings will be taken over the entire H-Q range. This will be sufficient data to evaluate the radial thrust at any flow.

The radial thrust can be calculated from the measured data as shown on sketch 400.9.110.394. {FIG. 1-16}. The pressure reading is multiplied by the circumferential area of the mpeller assigned to that tap. The resultant of the force vectors then represents the radial thrust. A more exact method is to carefully plot a pressure profile around the impeller. This profile is broken into convenient small increments and the vector sum represents the radial thrust.

INSTRUMENTATION

Instrument Type:	Dead weight gauge tester (Sulzer/Haenni System)
Range:	0.85 to 3000 ft. of water
Accuracy:	± 0.2 psi
Calibration:	Control of gauge piston dimensions "

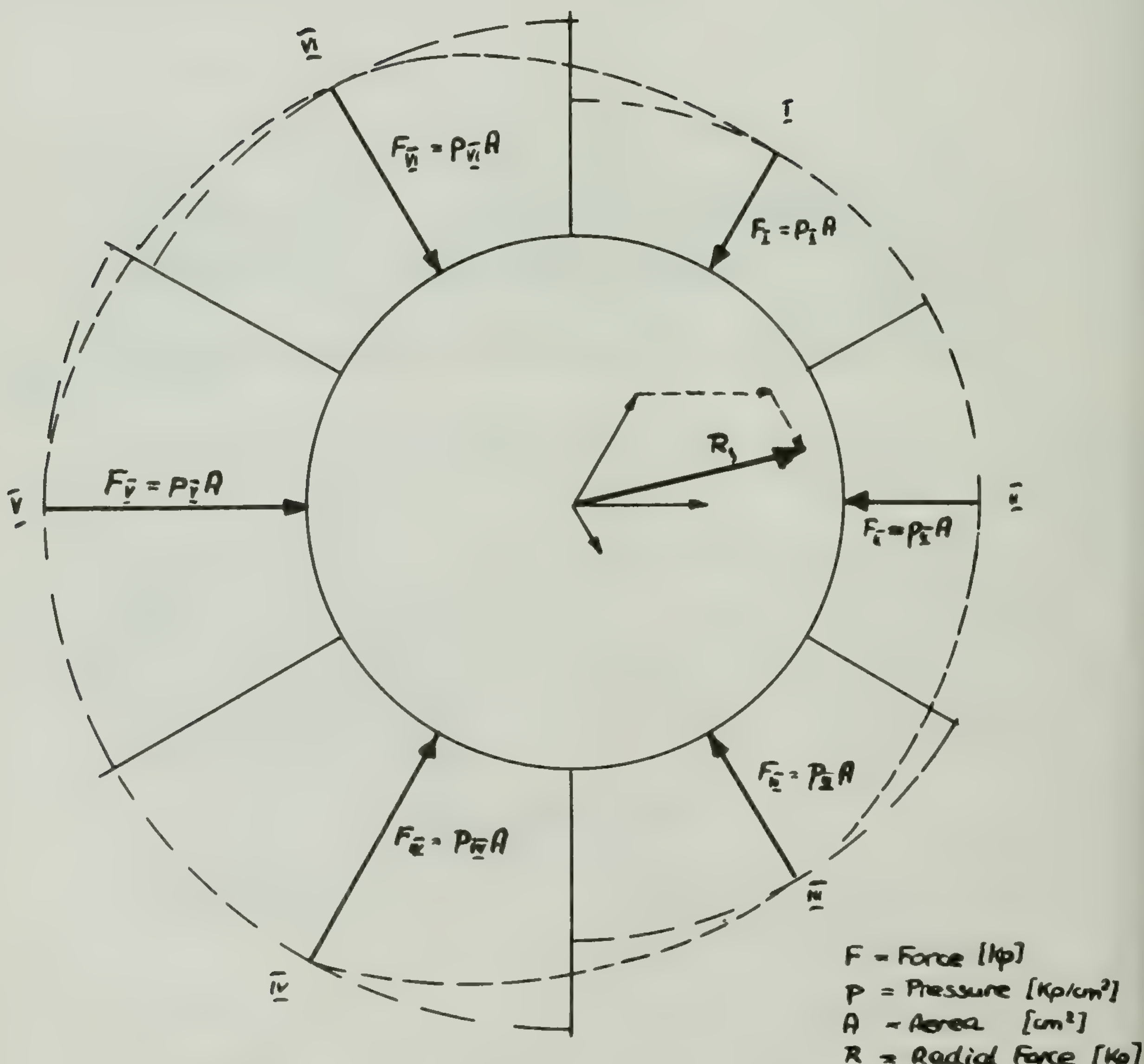
(End of Allis Chambers/Sulzer Test Procedures Reports)

Material prepared for this spot falls within the category of information considered proprietary by Daniel, Mann, Johnson, and Mendenhall and has been deleted at the request of that firm.

SULZER

TEHACHAPI PERT 33 ÷ 45/1

PRINCIPAL EVALUATION OF RADIAL THRUST BY SUPERPOSITION OF 6 (OR MORE) CIRCUMFERENTIAL FORCES.



400.9.110.39

FIG. 1-16

Nr. 42.00 0 - III 63 - 30 000

23 11 CA TL

2. B-L-H/Voith - Two-stage, Double Flow Model

a. Laboratory and Test Equipment Description

The following is a list and description of the test equipment and instrumentation to be employed while conducting the model performance tests:

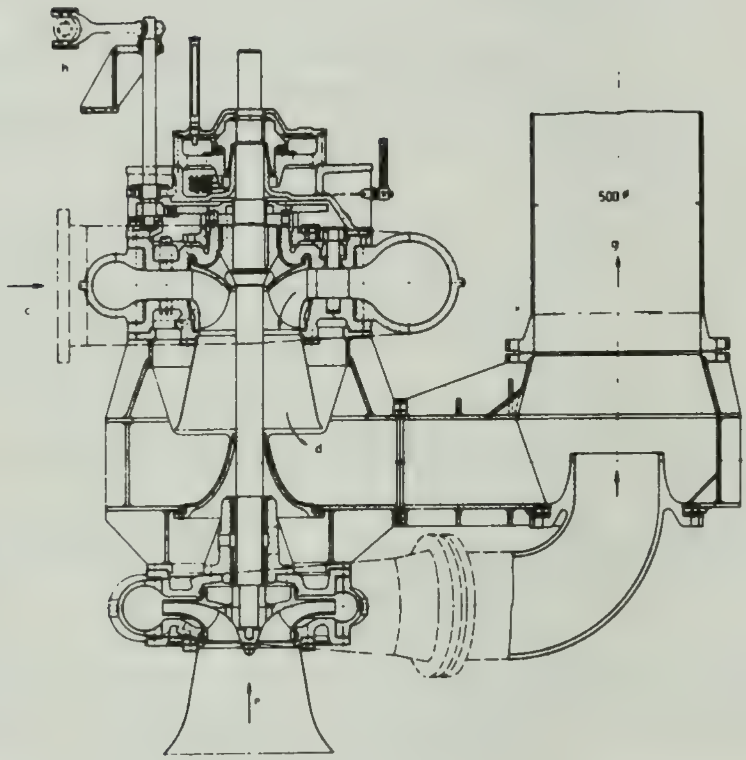
TEST EQUIPMENT DATA SHEET

TESTING FIRM:	J. M. VOITH, GMBH
Model Pump Driver:	
Type and Nameplate	Dynamometer built by J. M. Voith, GMBH
Data of Dynamometer	With 3-phase A.C. slip ring motor and 3-phase A.C. liquid rheostat.
Motor:	(Continental-Electroindustrie AG., Schorwerke)
Output (KW):	1500
Speed (RPM):	1485
Possible Speed Variation	Between 1485 and 750 rpm, stepless by variation of the liquid rheostat, gear between motor and pump, motor and gear supported in a cradle frame. We possess three gears for the following speed reduction ratios: 1485/2300, 1485/3300, 1485/4700 rpm.
Torque Measurement:	
Method & Instrument:	Weighting by weights and by dead weight tester measurement of the reastion torque of motor using home-built dynamometer
Range:	0 - 1500 KW
Accuracy:	± 0.05%
Method of Calibration	Calibration by calibrated measuring weights

Speed Measurement:	
Method & Instrument:	Electronic counter (developed by Voith)
Range:	0 - 600 rpm
Accuracy:	$\pm$ 0.02%
Method Calibration:	Check against basic standard.
Flow Capabilities:	
Maximum available flow as limited by:	
Piping	Pipe about 10" diameter for maximum flow of about 24 cfs
Measuring Canal	For weir measurement 1 x about 17.5 cfs, 2 x about 35 cfs
Booster Pump	Combined water turbine & centrifugal pump, called pressure reducer (home-made design). {FIGS. 1-17 and 1-18}
Discharge Measurement:	
Instruments:	1) Venturi meter 2) Weirs
Range:	up to 21 cfs / up to about 17.5 cfs or up to about 35 cfs
Accuracy:	At least $\pm$ 0.2% / At least $\pm$ 0.3%
Calibration:	Possible up to 21 cfs by calibrated tank
Control Measurement:	1) Main measurement Venturi meter 2) Control Measurement Weirs

<u>Head Capabilities</u> Maximum Permissible Pressure in Discharge Line:	2000 feet
Discharge Pressure Adjustable by:	Special throttling valve (our design diaphragm valve)
Discharge Pressure Measurement: Methods Used: Range & Accuracy: Calibration: Control Measurement:	 Dead weight tester (of our own design) 3280 feet. ± 0.05% Loading by calibration weights If necessary, second dead weight tester.
Maximum & Minimum Suction Pressure & Relationship to flow:	Adjustment by pressure reducer in the range of 33 - 164 feet Up to 21.2 cfs {See FIGS. 1-17 & 1-18}
Suction Pressure Adjustable by:	Guide vane adjustment of the turbine of the pressure reducer.
Positive Suction Pressure Applied by:	Guide vane adjustment of the turbine of the pressure reducer.
Suction Pressure: Method & Instrument Type: Range & Accuracy: Calibration:	 1) Dead Weight tester 2) Liquid manometer 3 - 328 feet ± 0.05% ; 1.5 - 65 feet ± 0.05% Loading by calibrated weights.

5



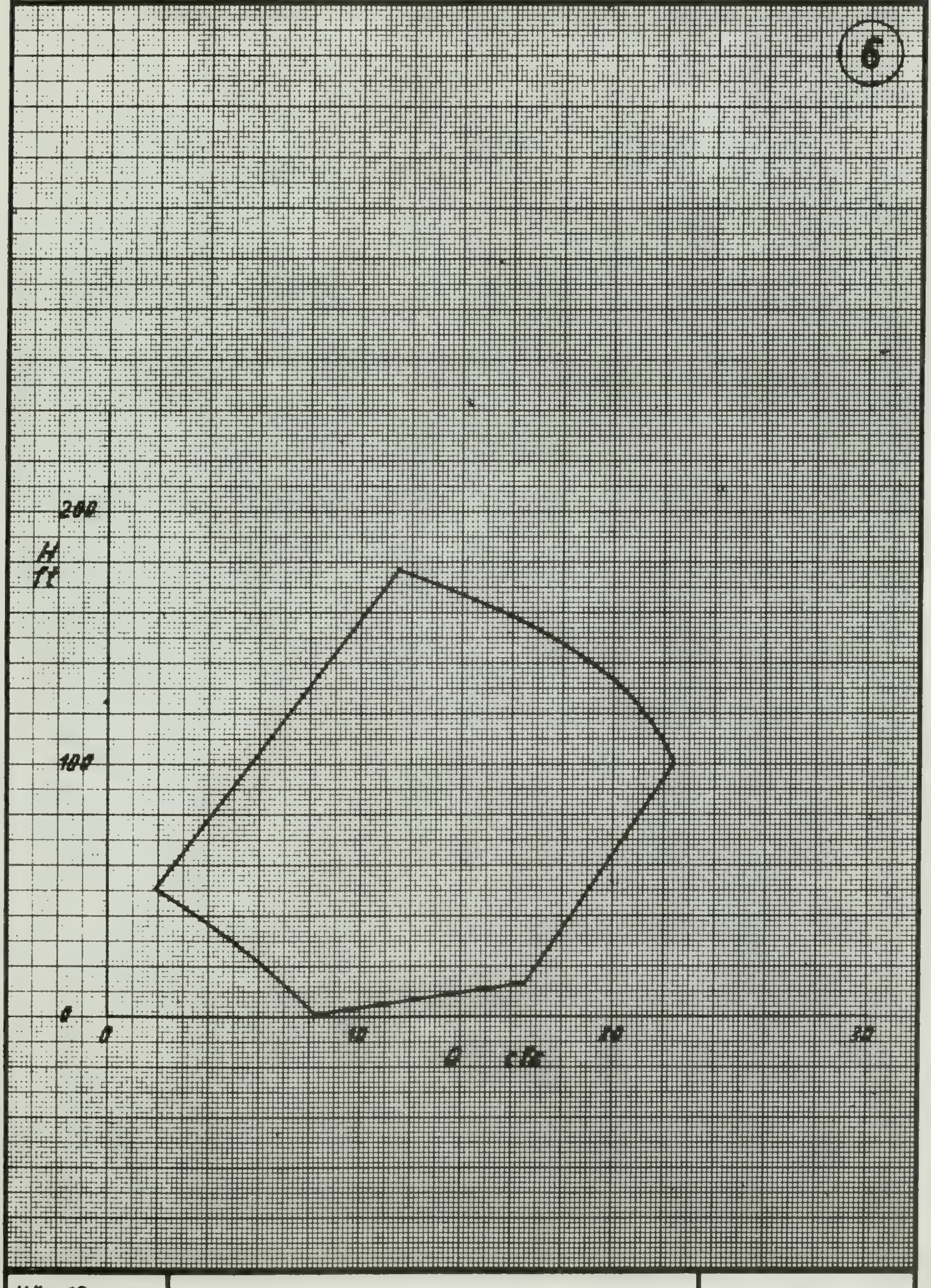
Pressure reducer,
Combination of Francis turbine and Centrifugal pump

a Francis turbine
b Centrifugal pump
c Water inlet from penstock
d Water outlet from turbine
e Water inlet in pump
f Water outlet from pump
g Water supply piping to model pump
h Wicket gate adjustment

FIG. 1-17

Pressure reducer
Operating range

Tehachapi



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VOITH

2. 82 -3719

FIG. 1-18

Voith has submitted the following document and drawings, discussing the test set-up and general pump performance test procedure:

"TEST SET-UP

The final layout of delivery and suction piping is shown on the test-arrangement Drawing 2.83-8498 {Vol. I, Ch. 7}. The suction piping, which is run alongside the machines, has an inside diameter of 15.7 in. This suction pipe directs the water first vertically upwards, then to the test stand and eventually from above through the suction bend to the model pump. The drawing shows the 1.5 MW driving motor. On one side, the double-flow, double-stage model pump is connected and on the other, by way of example, a single-flow, single-stage model pump. The inside diameter of the pump suction bend is 11.8 in. The associated Y-pipe, as may be noted from Drawing 2.83-8541 {Vol. I, Ch. 7} has an inside connection diameter of 19.7 in. Therefore, ahead of the Y-pipe a taper pipe (diffuser), 15.7/19.7 in. diameter, must be fitted, the angle formed by the horizontal and the taper being about 5^{o}. In order to obtain a uniform inflow to the taper pipe; this pipe is preceded by a 90^{o} bend with guide plates, inside diameter 15.7 in. The delivery pipe leading to the different pump test stands runs straight, inside diameter 9.8 in. One the right-hand side, the pipe is connected which runs to the reservoir for storage and calibration. This pipe also supplies the water for turbine operation. In order to measure the turbine water flow, a symmetric Venturi tube (B) will be used, as shown on Drawing 2.83-8389 {FIG. 1-19}. From the pump delivery connections the water is directed to the high-pressure measuring pipe via 2 Tees. These Tees have a connection inside diameter of 9.8 in.; in the zone where the water is re-directed the Tee has an inside diameter of 11.8 in. From these Tees the water flows through the delivery pipe to the Venturi tube (A). For measuring, Venturi tubes of the Herschel type have been chosen, which are also shown on the above-mentioned Drawing 2.83-8389 {FIG. 1-19}. The straight gauging section ahead of this Venturi tube is about 22 x pipe diameter. Behind the Venturi tube another straight pipe of about 9 x pipe diameter will be provided. The dimensions of the Venturi tube (A) substantially comply with the specifications of the Standards of the Hydraulic Institute. The measuring pressure ahead of the Venturi tube is taken off at point "a". The measuring pressure in the narrowest cross section can be taken off at "b" and "c". Point "c" roughly corresponds to the point specified in the Standards of the Hydraulic Institute.

At point "b" the flow is still slightly accelerated. We hope that we shall obtain a constant pressure indication without major pressure variations. All parts of the Venturi tubes in contact with the water are made of stainless steel. The Venturi tubes had to be manufactured in our works, as for such abnormal dimensions no German manufacturer could be found in view of the

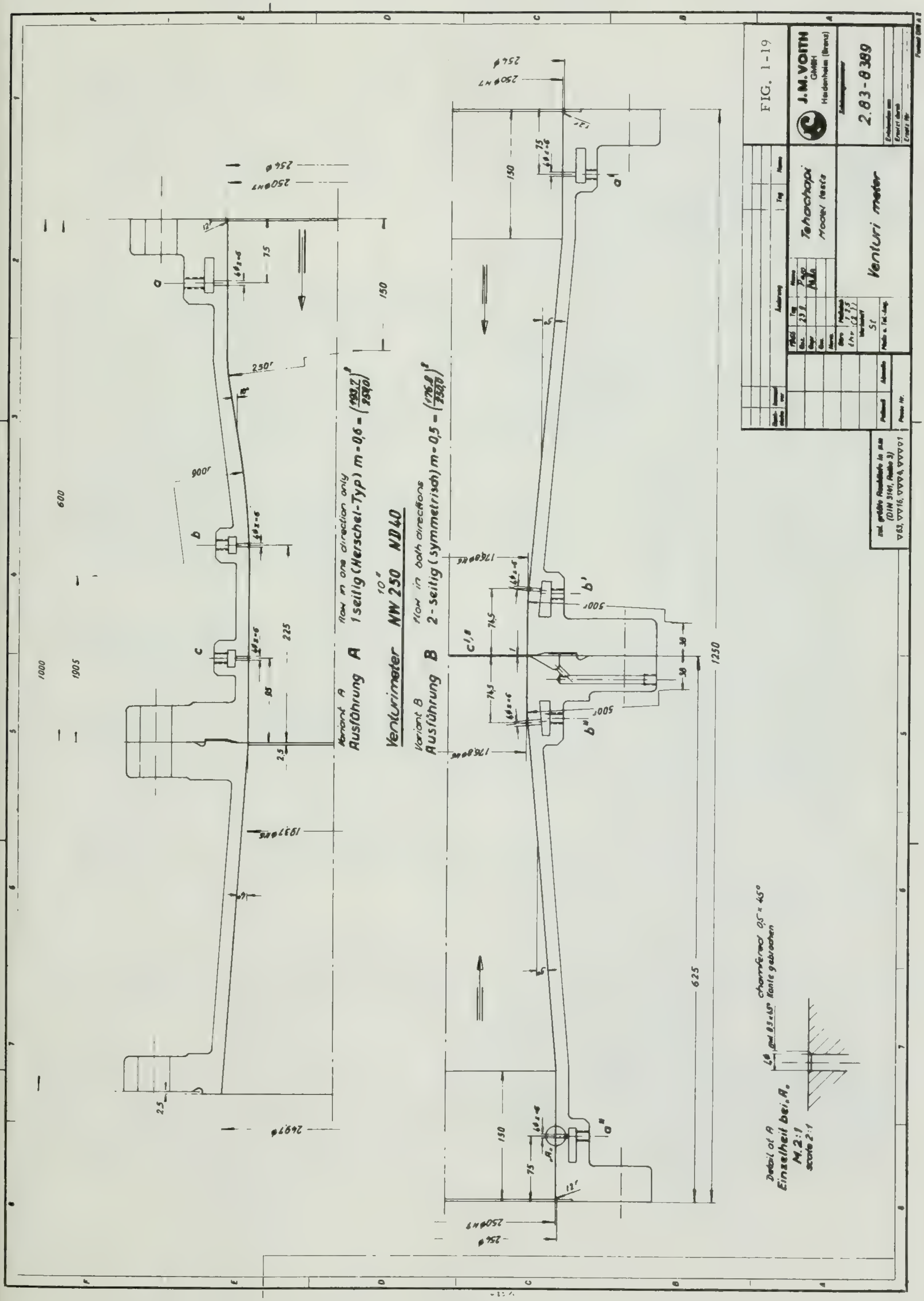

Venturimeter NW 250 ND 40
Variant A
Ausführung A
flow in one direction only
1 seilig (Herschel-Typ)
Variant B
Ausführung B
flow in both directions
2-seilig (Symmetrisch)
J.M. VOITH GmbH Heidenheim (Brenz)
2.83-8389
Venturi meter
Detail at A
Einzelheit bei A
M. 2:1
scale 2:1
chamfered 0,5 × 45°
1250
625
600
1000
1905

FIG. 1-19

short time available. At the end of the long delivery pipe a throttling valve is provided which is to dissipate the energy of the water without vibrations and with a minimum of noise and to direct the water to the inlet of the measuring channel. A weir at the end of the measuring channel is used as second water measuring equipment.

"Calibration of the Venturi Tubes

The Venturi tubes (A) and (B) and the weir in the measuring channel will be calibrated simultaneously. For the calibration a high-level tank will be used (approximate diameter 118 ft., 26 ft. deep). From the high-level tank, in an accurately determined period of time t_L (sec.), an accurately measured water volume V (m^3) is removed and directed to the Venturi tubes and the measuring weir. The readings of these measuring equipments (effective pressure and head above weir crest) are recorded; they correspond to the water flow Q = V/t (m^3/sec.). The water volume V is determined from the measured diameter of the high-level tank and from the draw-off of the water level in this tank. The diameter of the high-level tank is measured at several cross-sectional areas (up to 13 ft. deep) and at 8 points ot the periphery.

The last time the diameters of the high-level tank were determined was in 1954. To our regret, it is not easy to repeat the checks of these diameters. We have therefore requested the surveyor of the City of Heidenheim to check geodetically on the high-level tank. This work has been completed in the meantime. However, the report and the translation of this report are not yet available so that we must ask you to have still some patience.

{The report of this work and the calibration of Venturi meters has since been reviewed and witnessed by DMJM representatives.}

"Electric Motor

The model pumps will be driven by a three-phase A. C. slip ring motor with two shaft ends (Drawings 2.82-4152/53/54). {FIG. 1-20, 1-21, and 1-22}

The motor is rated at 1,500 KW at 1,485 RPM. For the measurement of the reaction moment, the frame of the motor is supported by flanged-on hollow bearing journals at either end so as to be capable of rocking. Hydrostatic bearings will be provided to increase the accuracy and sensitivity of the torque measurements.

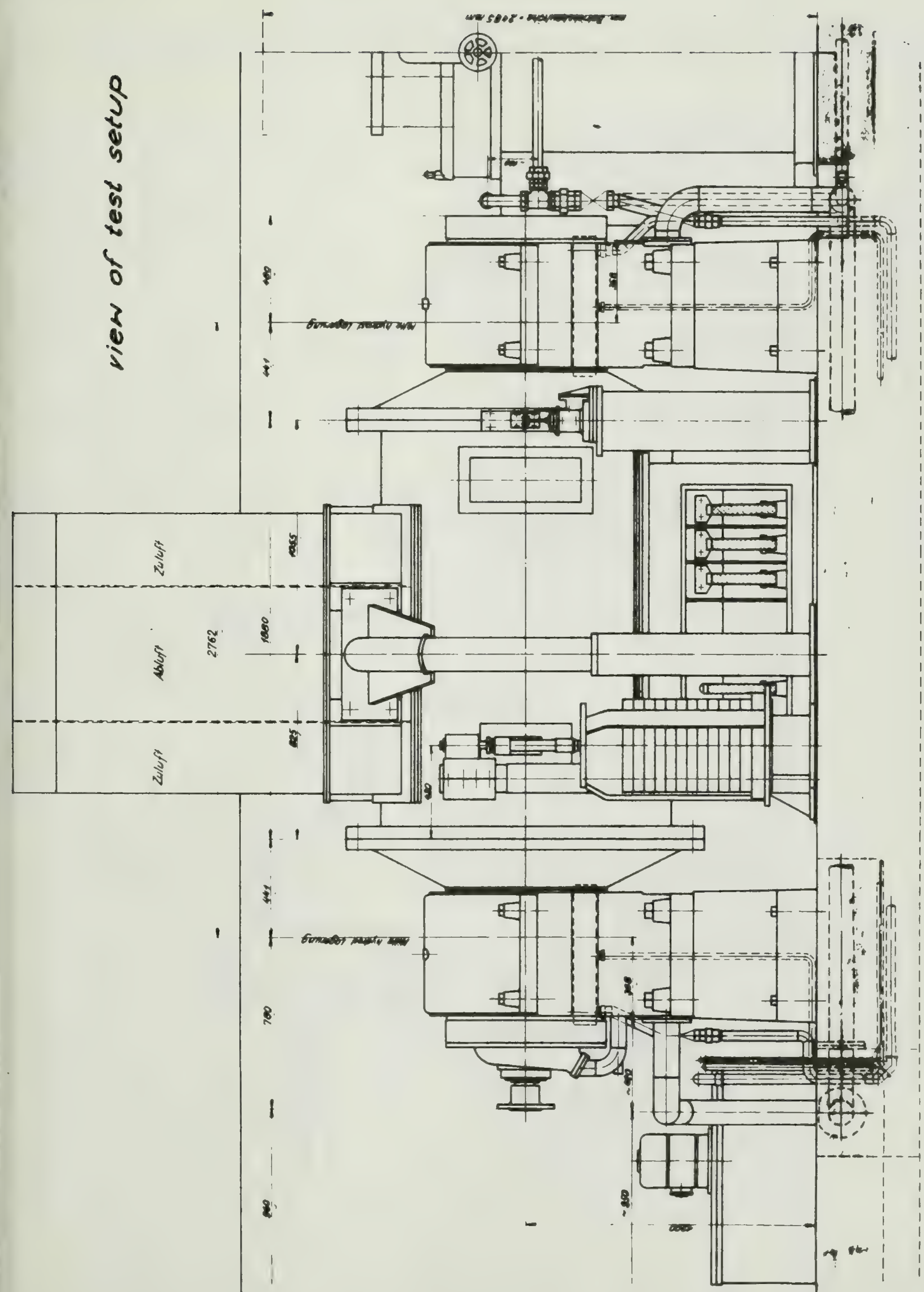

view of test setup
Zuluft
Abluft
Zuluft
2762
1880
1065
825
441
441
780
840

FIG. 1-20

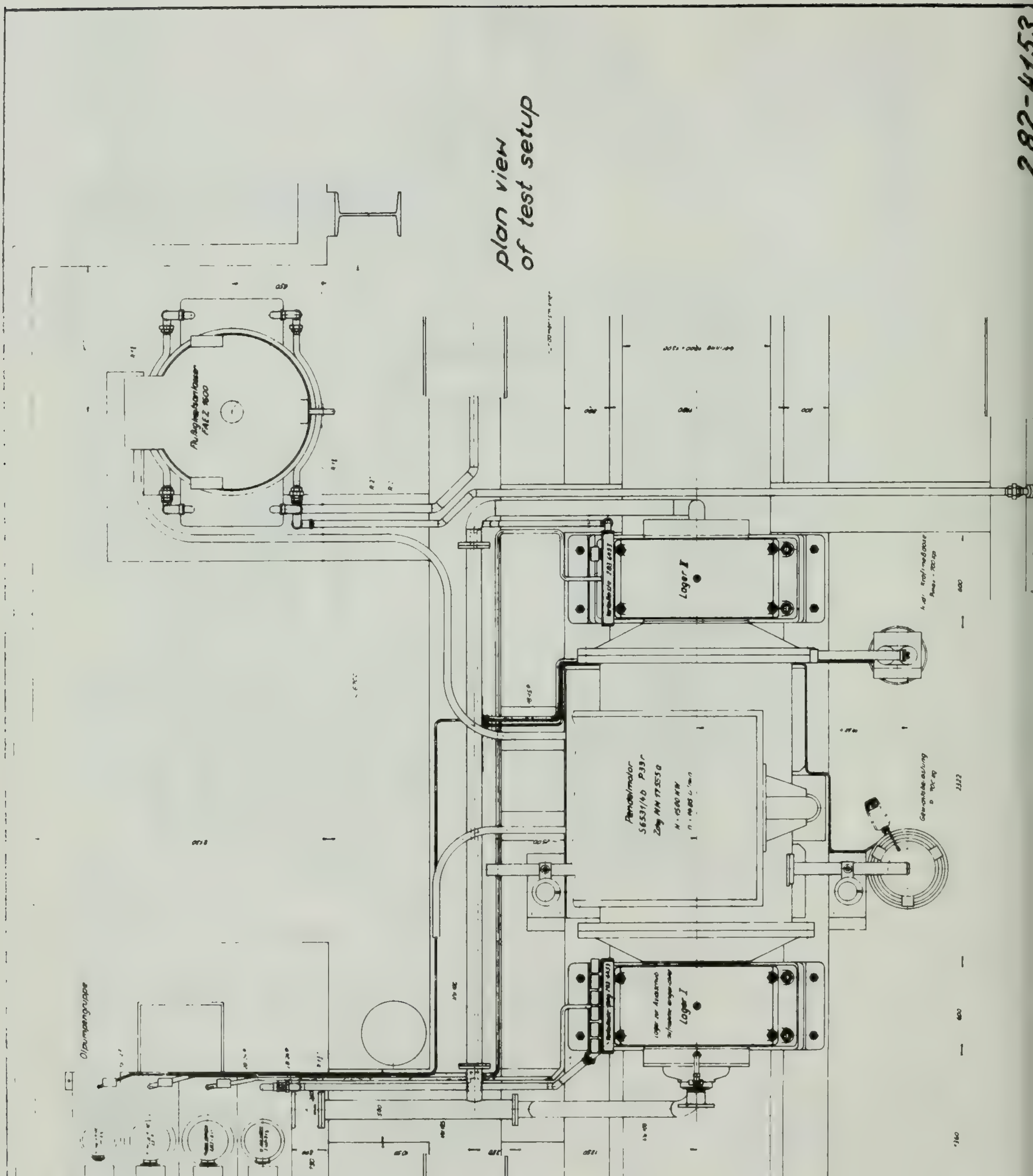


FIG. 1-21

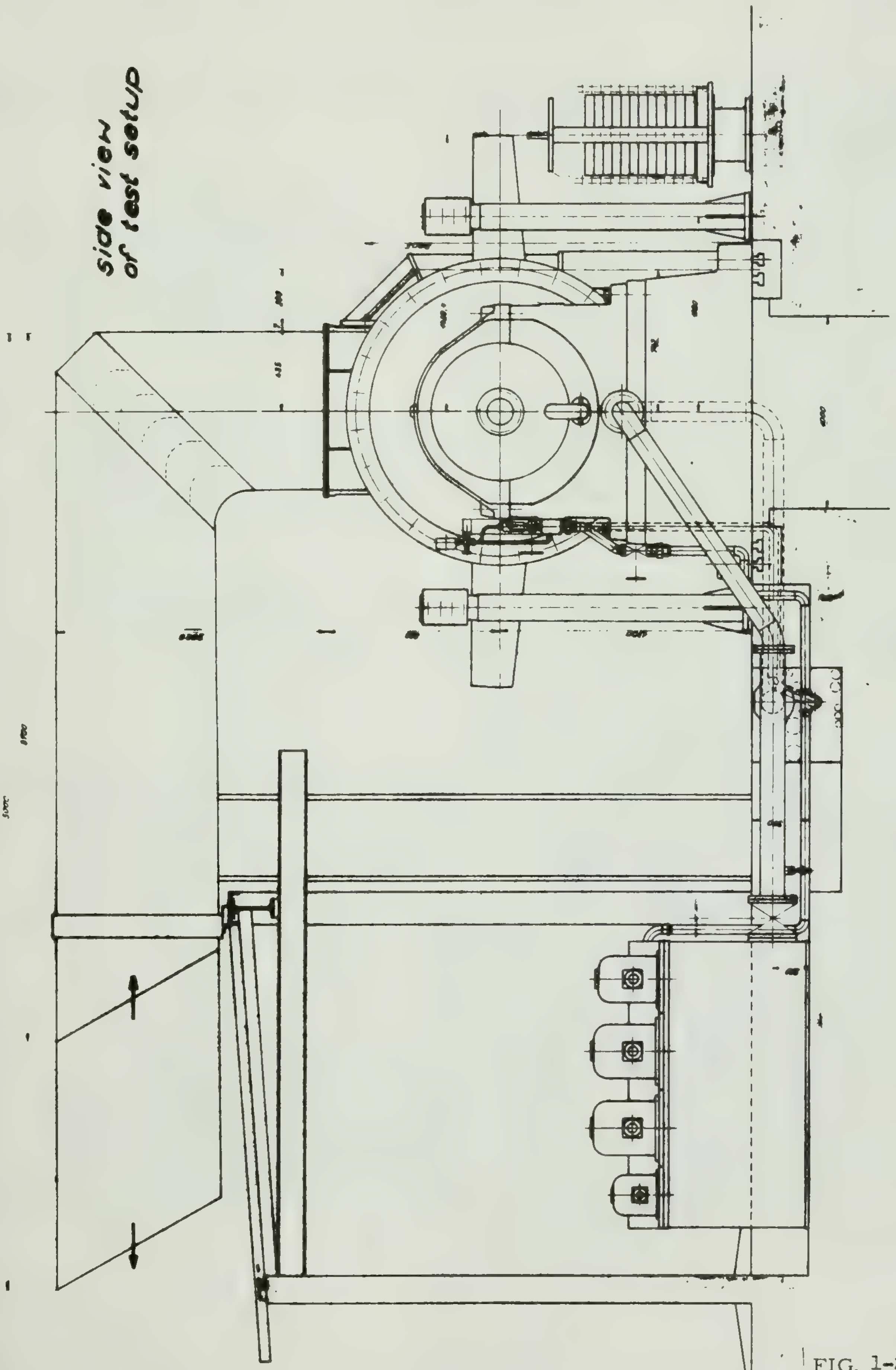


FIG. 1-22

The motor speed can steplessly be varied between 1,485 and 750 rpm. This is achieved by a three-phase A. C. liquid rheostat by means of which the resistance of the rotor circuit is varied.

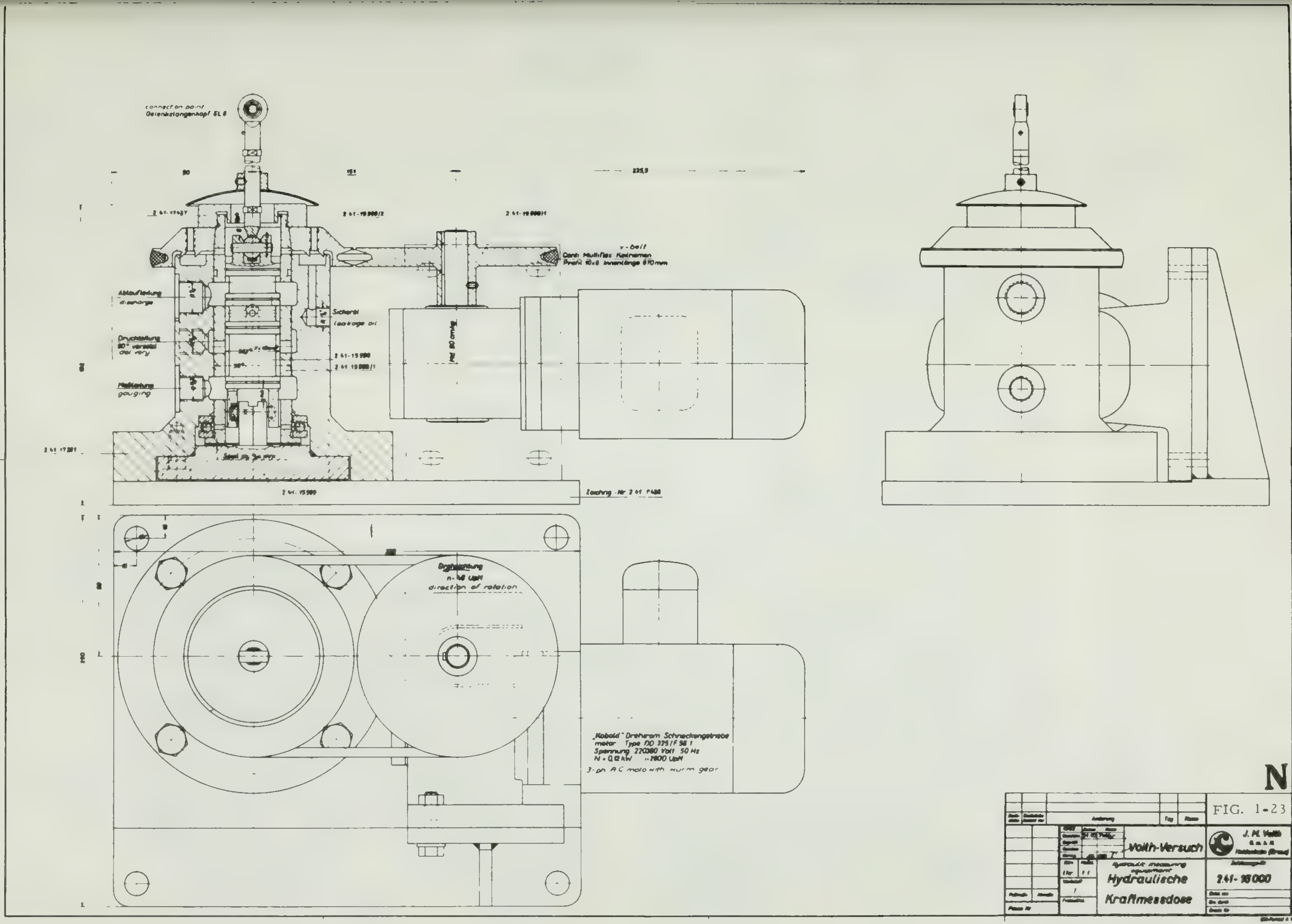
The motor operates at 6,000 volts. The motor is connected to the grid and the liquid rheostat to the motor via extremely flexible copper tapes. The influence of these copper tapes on the torque measurement will be considered by calibration. The electric connections are arranged underneath the stator casing in a plate box physically separated from the stator casing. The flexible copper tapes are so fitted that they can be observed through a plexiglass observation window.

A gear is interposed between electric motor and model pump in order to obtain the required model pump speeds. For this application we use planetary gears of our own design and manufacture. These planetary gears are rigidly attached to the motor casing which is suspended in a cradle-mounted frame.

The 3 gears provide speeds of 4,700, 3,300 and 2,300 rpm. At these speeds the full motor output is available. If speed regulation by means of the liquid rheostats is adopted, the power of the motor, as is generally known, decreases with increasing speed.

"Torque Measurement

As already mentioned above, the torque of the model pump can be measured by the cradle-frame mounted motor casing. The gear is rigidly connected to the motor casing. The losses in the gear will in this way be eliminated the same way as the losses in the motor. The reaction torque of the motor, which in magnitude is identical with the torque of the model pump, is computed from the force accurately measured at a lever arm and from the measured length of this lever arm. The bulk of the force is determined by means of dead weights which have been calibrated by the state authorities. The residual force is measured by means of a second lever arm and a hydraulic measuring equipment, which is shown on Drawing 2.41-16000 {FIG. 1-23}. This measuring equipment substantially comprises a revolving piston (diameter 1.405 in., area 1.551 sq. in.). The oil pressure which is created underneath the piston is a measure for the residual force to be measured. The oil pressure is measured by a piston gauge. The equipment can be calibrated by placing calibrated dead weights on the platform and reading off the indicator of the piston gauge. The equipment is so designed that it can automatically hold the cradle-frame mounted motor in "zero" position.


connection point
Gelenkstangenkopf GL 8
v-belt
Conti Multiflex Keilriemen
Profil 10x6 Innenlänge 670mm
Ablaufleitung
discharge
Sickeröl
leakage oil
Druckleitung
90° versetzt
del very
Meßleitung
gauging
Drehrichtung
direction of rotation
„Kobold" Drehstrom Schneckengetriebe
motor Type DD 325/F 58 1
Spannung 220/380 Volt 50 Hz
3-ph A.C. motor with worm gear
Voith-Versuch
J. M. Voith
hydraulic measuring equipment
Hydraulische
Kraftmessdose
2.41-18000
FIG. 1-23

"Speed Measurement

Electronic counters of Voith design and manufacture will be used. Both the speed at the input and at the output end of the gear can be measured by counters.

"Determination of the Delivery Head

Piston gages of our own design and construction, which can be loaded by dead weights and can be calibrated, are used to determine the delivery head. {FIG. 1-24} shows a section through this equipment and the general arrangement. In a stationary cylinder a revolving piston is free to glide, which on the one side is fitted with a dead weight platform. Additionally, the platform acts on a stationary coil spring, the compression of which is a measure of the residual force which exceeds the dead weight load. In this way, the height setting of the dead weight platform is a measure of the residual force. The height of the dead weight platform is determined by a reaction-free feeler and indicated on a scale. Pressures below 6.5 ft. w.g., are measured by U-tube manometers. Bourdon-type manometers are used for checks.

"Determination of the Water Flow

The water flow is measured by Venturi tube (A) as mentioned under 7a). This required a measurement of the differential pressures between the points "a", "c" and "a", "b" respectively. For the measurement of the differential pressure mercury gages of Messrs. Bopp & Reuther, Mannhein and of the Debro-Werke, Düsseldorf, will be used. Such measuring equipment is standard in our laboratory. The measuring equipment is suitable up to a rated pressure of 853 psig, or in other words, up to absolute pressures of 908 psia and for differential pressures of 908 psia and for differential pressures up to 6.6 ft. of mercury.

"Water Quality

For the conduct of the tests the water is taken from a large natural spring The water temperature is constant at 48 - 50° F. The water is clean and free from any admixtures. The air content is determined by an instrument of the Shyke type. A new instrument of the same type has been ordered in Great Britain. The water from the spring is delivered by pumps at the desired supply pressure of the model pump and flows then through the suction pipe, inside diameter 15.7 in. to the model pumps.."

(END OF J. M. VOITH REPORT)

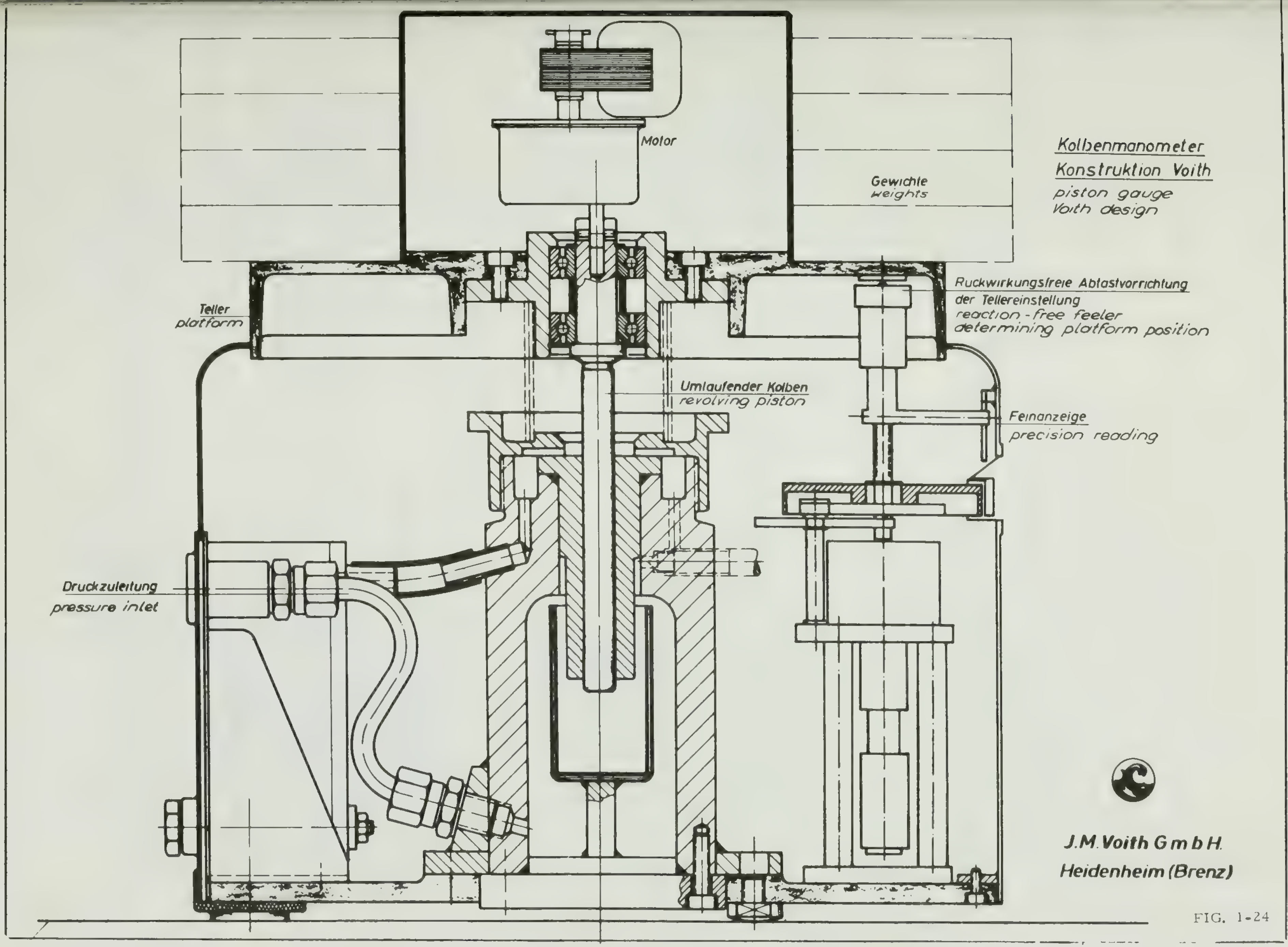

Kolbenmanometer
Konstruktion Voith
piston gauge
Voith design
Motor
Gewichte
weights
Teller
platform
Ruckwirkungsfreie Abtastvorrichtung
der Tellereinstellung
reaction-free feeler
determining platform position
Umlaufender Kolben
revolving piston
Feinanzeige
precision reading
Druckzuleitung
pressure inlet
J.M. Voith GmbH.
Heidenheim (Brenz)

FIG. 1-24

b. Model Pump Test Procedures

J. M. Voith, GMBH, has submitted the following test procedures, describing generally the following required tests:

1) Pumping characteristics

2) Braking and Turbining characteristics

3) Conduct of the cavitation tests

4) Determination of the symmetry of the discharge from the pump impeller in the case of the single-flow, single-stage pump.

5) Measurement of the shaft deflections of the double-flow two-stage model pump.

6) Determination of the prerotation

7) Measurement of the pressure variations.

"J. M. VOITH TEST PROCEDURE

"Pumping Characteristics

As usual, the delivery Q, the delivery head H, the torque M and the speed n of the pump are measured. From these values the efficiency is determined. The values measured at a given speed are converted to the agreed reference speed and shown in graphs. From the measured gross pump input the mechanical losses of the pump shaft, determined in separate test runs, are deducted.

"Braking and Turbining Characteristics

For the study of the water hammer, which occurs at the pumps in the event of power failure, the machines must also be investigated during turbining and braking. For this mode of operation the water must flow through the machine in the opposite direction, thus from the delivery branch to the suction branch. The required pressure water is taken from the high-level tank, the water flow through the Venturi tube, model B, is measured, and then the water passes through the model pump and the suction pipe (15.7 in inside diameter) via a throttling valve into the tailwater. The water pressure ahead of the machine will be about 260 - 300 feet for braking, the normal direction of rotation of the machine is retained which, in conformity with the lower operating pressure, runs at a lower speed. These operating conditions can readily be accommodated with the dynamometer (driving motor 1.5 MW) installed in our laboratory. For turbining, the machine runs in the opposite direction to that for pumping. Therefore, it will be necessary to brake the machine. Theoretically it should be possible to use, for this application also, the driving motor for pumping if a gear with a higher transmission ratio is interposed. Should this cause difficulties, the model pump will be disconnected from the driving motor, and at the other shaft end a hydrodynamic Froude brake will be attached, which, because of the use of a cradle-mounting, provides a satisfactorily accurate measurement of the developed torque. A drawing showing the general arrangement of this brake will also follow. For the measurement of the delivery, delivery head and speed values we would use the same measuring equipment as for the normal determination of the pump characteristics. All measured values will be referred to the reference speed.

"Conduct of the Cavitation Tests

For the cavitation tests the inlet head of the pumps is varied. With approximately the same adjustment of the throttling valve in the delivery pipe, for each operating point different inlet heads are obtained by adjustment of the feeder pump, so that different σ values (NPSH values) are obtained. For each measuring point, as during the main tests, delivery, delivery head, torque and speed are measured. Furthermore, for each measuring point the cavitation on the suction side of the vanes is observed.

Some of the operating conditions will be recorded on photos to the extent that the gas content of the water permits making such photos.

The change of the delivery head and the change of the efficiency at approximately the same testing point as a function of the Thoma cavitation coefficient σ or of NPSH will be shown in graphical form.

"Determination of the Symmetry of the Discharge from the Pump Impeller in the Case of the Single-Flow, Single-Stage Pump

This test will be run only upon the special request of the customer. For this test the wall pressure in the transition zone between impeller outlet and guide vane inlet will be measured by 13 bores at the same diameter spread uniformly between the vane pitch of the guide vanes. The connections of the gauging bores must be led outwards and the pressure measured by means of U-tube manometers. The tests will be conducted for the different operating conditions of the pump and are intended to provide evidence on the radial loads on the impeller at different operating conditions.

"Measurement of the Shaft Deflection of the Double-Flow, Two-Stage Model Pump

Also these measurements will be made only upon the special request of the customer. For these tests, as may be noted from the attached Drawing 2.83-8528, {FIG. 1-25} in the guide vane ring of the second stage of the pump, induction type transmitters are arranged at four different points evenly distributed over the periphery. These transmitters, via a carrier frequency bridge, permit measurement of the displacement of the impeller, and thus of the shaft, with the equipment in operation, without the necessity of any physical contact. The measured values are recorded by an oscillograph.

"Determination of the Prerotation

With the chosen feed arrangement, neither for the single-flow, single-stage nor for the double-flow, double-stage pump we anticipate an influence of the prerotation on the pump characteristics. In the suction bend, additional ribs are provided. In order to provide evidence that there is no prerotation, during the additional or preliminary tests the direction of flow will be determined by small flags which will be arranged at several points at the inlet of the suction bend. The attached Drawing 2.83-8527 {FIG. 1-26} shows the general arrangement which we propose to use. The position of the flags can be observed from the outside through plexiglass windows fitted subsequently on the suction bends also on the double-flow, double-stage pump.

"Measurement of the Pressure Variations

At the spiral case of the pump, immediately ahead of the flange at the spiral outlet, a hydraulic measuring equipment is fitted. The measured pressure is recorded by an oscillograph. In this way, the pressure variations inside the pump can be determined. The instrument will be calibrated before and after the tests."

(END OF J. M. VOITH REPORT)

Material prepared for this spot falls within the category of information considered proprietary by Daniel, Mann, Johnson, and Mendenhall and has been deleted at the request of that firm.

3. Byron Jackson - Single-stage Model

a. Equipment

TEST EQUIPMENT DATA SHEET

TESTING FIRM:	BYRON JACKSON PUMPS, INC.
Model Pump Driver 1	
Type and Nameplate	General Electric Induction Motor
Data of Dynamometer	1750 HP 2 Pole 4160 Volts 3-phase 60 cycle with Reduction Gear.
Output (KW):	1300 Kilowatts
Speed (RPM):	Full Load Speed 3565 RPM
Possible Speed	None Required
Variation	Speed as determined by motor speed and appropriate gear ratio.
Model Pump Driver 2	
Type and Nameplate	General Electric D. C. Electric Dynamometer
Data of Dynamometer	Model 265G81 250 volts 1100 Amps. Absorbes 400 HP, Delivers 300 HP 1500 to 4000 RPM Torque Arm - 25.210". This Dynamometer will be used for three (3) quadrant tests.
Output (KW):	As Required
Speed (RPM):	As required, variable from 1500 to 4000 RPM.

Possible Speed Variation	As required from 1500 to 4000 RPM Speed control with Boiler & Chivens Speed Controller
Torque Measurement:	
Method and Instrument:	Lever arm and known calibrated weight standards.
Type:	Direct readout calibrated beam scale.
Range:	As Required
Accuracy:	Overall accuracy 0. 1%
Method of Calibration:	Dead Weight Standards
Speed Measurement:	
Method & Instrument:	Boller-Chivens Speed Controller and Hewlett Packard Electronic Counter Model 521 c.
Type:	Counting output signal in CPS from a 60-tooth sprocket mounted on pump shaft.
Range:	Range 1/60 sec. to 1667 Sec.
Accuracy:	0.03% + 1 RPM when count is one (1) Sec.
Method of Calibration:	Internal built-in counting of time base

<u>Model Pump Driver 3</u> Type and Nameplate Data of Dynamometer Motor	General Electric Induction Motor 300 HP, 4 pole, 2200 volts, 3-phase Motor No. 4047879 This motor will be used in preliminary characteristic studies
Output (KW):	225 Kilowatts
Speed (RPM):	Full load speed - 1750 RPM
Possible Speed Variation	None Required
Torque Measurement: Method and Instrument: Type: Range: Accuracy: Method of Calibration:	 General Electric or Weston Laboratory Type calibrated polyphase wattmeter As required Overall Accuracy 0.25% With known electrical standards, together with appropriate current & potential transformers.

Speed Measurement:	
Method and Instrument Type:	Hewlett Packard Electronic Counter-Model 521c counting output signal in CPS from a 60-tooth sprocket mounted on the pump shaft.
Range:	1/60 sec. to 1667 sec. - using 60 cps
Accuracy:	0.03% + 1 RPM when counting one (1) Sec.
Method of Calibration:	Internal counting of time base frequency - built in
Torque Measurement:	
Method & Instrument Type:	Torque meter Lebow and Associates Model 234-48K with Cox Digital Readout.
Range:	48,000 inch-pounds
Accuracy:	Overall accuracy 0.1%
Method of Calibration:	Lever Arm on known standard calibrated weights.
Speed Measurement:	
Method & Instrument Type:	Hewlett Packard Electronic Counter Model 521c counting output signal in CPS from a 60-tooth sprocket mounted on the pump shaft.
Range:	1/60 sec. to 1667 sec. using 60 cps
Accuracy:	0.03% + 1 RPM when count is one (1) Sec.
Method of Calibration:	Internal counting of time base frequency built in

Flow Capabilities	
Maximum available flow as limited by:	Not limited within scope of described tests.
Flow Measurement:	Herschel Type Venturi Meters with differential water manometers.
Range:	40 feet water
Accuracy:	Overall accuracy 0.1%
Calibration:	Volumetric calibration tank
Control Measurement	Small Standard Volumns
Head Capabilities	
Maximum Permissible Pressure in Discharge Line:	Will be designed for maximum requirement.
Discharge Pressure: Adjusted by:	Throttling plug valve with possible addition of friction tubes if necessary.
Discharge Pressure Measurement:	
Instrument Types:	Mercury column, Bourdon tube gages and a Differential Mercury Manometer.
Range & Accuracy:	Full range as required. Hg. Col. - 0.03% Bourdon gages 0.5%.
Calibration:	Dead Weight Tester
Control Measurement:	Standards of weights and measures. Head can be measured with a Differential Mercury Manometer.

Maximum & Minimum Suction Pressure and Relationship to Flow:	
Suction Pressure: Adjustable by:	Valves in loop injection & outlet lines.
Positive Suction Pressure Applied by:	Injection pump to loop
Suction Pressure Measurement:	
Method and Instrument Type:	Mercury column 0 - 15. 5 ft.
Range & Accuracy:	0-15. 5 ft. - Accuracy 0. 03%
Calibration Control Measurement:	As required. Standards of weights of measures

b. Model Pump Test Procedure

Byron Jackson has submitted a test procedure, listing the test equipment and instrumentation to be employed for the normal pump phase of the model pump tests. A copy of this procedure, "Preliminary Test Loop Design and Test Procedure - Normal Pump", Report 3c, follows:

August 14, 1964
Revised September 24, 1964

"TEHACHAPI PUMPING FACILITY

DMJM SPECIFICATION NO. 637-1-1C

Ref: Section 4 - Preliminary Test Loop Design and Test Procedure - Normal Pump

Report No. 3c

"1.0 REFERENCES:

1.1 DMJM Specification 637-1-1C dated 4-15-64.

1.2 ASME Power Test Code for Centrifugal Pumps - DTC.8.1 - dated 1954.

1.3 Standards of Hydraulic Institute, Centrifugal Pump Section - dated 1951, Rev.

1.4 Byron Jackson Division - Proposal for the Design, Fabrication and Testing of Pump Models for the Tehachapi Pumping Plant - dated February, 1964.

"2.0 PURPOSE:

The purpose of this document is to present the intended procedures and general conduct of the normal pump tests as set forth in the referenced specifications.

"3.0 DESCRIPTION OF TEST SPECIMEN:

Pump - A Byron Jackson Type 10 x 10 x 16 single-suction diffuser pump. Design specific speed $N_s = 2000$.

"4.0 DESCRIPTION OF THE MODEL TEST STAND WITH INSTRUMENT DATA:

4.1 Drawing 1C-2430 {Vol. I, Ch. 8} "Test Stand - Tehachapi Model Pump" shows a detailed layout of the test stand.

4.1.1 Model Driver - a General Electric induction test motor. 1750 HP, 2 pole, 3-phase, 60-cycle, 410 volt - Serial No: 6879569.

4.1.2 Speed Reducer - a Western Gear Products speed reducer.

Ratio: 1.2476/1.0 Rating: 6400 HP

Serial No: 140

4.1.3 Booster Pump - a Byron Jackson vertical, turbine type circulator pump. Size: 28 RX.

A variable speed driver will provide controlled suction pressure to the model and adequate flow and temperature control through the course of the testing.

4.1.4 Flow Meter - a Herschel type venturi meter with an inlet to throat diameter ratio of 2:1.

Range: 0-10000 GPM.

4.1.5 Calibration Stand for in situ flow meter calibration - a 10,000 GPM calibration tank on a suspension lever scale connected to a timer.

4.1.6 Test Tank - a 400,000 GPM capacity reservoir.

4.1.7 All inter connecting piping, fittings and valves will be sized and located in accordance with the specifications under 1.0 and consistent with sound engineering practices.

4.2 Instrumentation and Data

4.2.1 Torque Meter - a Lebow Model 1234048K Rotary Shaft Torque Sensor, with pedestal mounting plate, speed sensor (pulse generator) air operated brush lifter, in situ calibrated with lever arm and weights to a Cox Model 522-1A Digital Torque Indicator for an 0.1% system accuracy.

Capacity: 48,000 in-lbs.

4.2.2 Calibration Tank Scale - a Fairbanks-Morse Double Link Suspension Cast Pipe Lever Scale on Stands, with a type registering beam and counterpoise. Beam graduated 1000 lbs. x 5 lbs. insitu calibrated to a system accuracy of 1/40 of 1%.

4.2.3 Pressure Tester - a Barnet Combined Hi-Low Pressure Dead Weight Tester, type 4140/1.

Low Pressure range 10-400 PSI
High Pressure range 400-4000 PSI
All calibration standards traceable to NBL.

4.2.4 Timer - a Hewlett-Packard Electronic Counter, crystal controlled.

Model 5214L

Accuracy: 2/100 of 1%.

4.2.5 Mercury Manometer - for accurate mensuration of model discharge pressure.

4.2.6 Water Manometer - for accurate mensuration of model suction pressure.

4.2.7 Differential Water Manometer - for reading flow meter differential values.

4.2.8 Thermocouples and Leeds & Northrup Galvanometer, Model 8657 - Serial No. 1321424 - for measuring pumping temperature at critical points in the test line.

"5.0 <u>PROCEDURE:</u>

5.1 INSITU Calibration of Venturi Meter:

After the erection of the test stand and before the actual tests on the model specimen, the Venturi type flow meter will be calibrated as required in referenced specification. This calibration will be accomplished by verifying the metered flow with weight measurements. At each flow point the following values will be recorded. Liquid temperature (as measured at the Venturi Inlet) in deg. F - manometer deflection (in inches of water), tare weight, final weight and time.

5.2 All instruments used will be properly calibrated before tests and as deemed necessary thereafter. Calibrations to be performed by Byron Jackson personnel or others.

5.3 Sufficient data shall be recorded to establish parameters of H.Q. from cut off to point of significant cavitation, of brake horsepower required by the test specimen, of the calculated efficiency and of the NPSH required as determined by actual tests.

5.4 Preliminary Study:

5.4.1 Preliminary tests will be made on two (2) separate impellers, and diffuser designs - designated as Impeller A - Diffuser A, Impeller B - Diffuser B.

5.4.2 The preliminary studies will require four tests - covering the following combinations:

Impeller A - Diffuser A
Impeller B - Diffuser A
Impeller B - Diffuser B
Impeller A - Diffuser B

The elevation of the data obtained in the above tests will result in the selection of the most satisfactory impeller diffuser combination.

5.5 H.Q. Efficiency Characteristics Tests:

5.5.1 Just prior to these characteristics studies, it will be necessary to recalibrate all instruments and meters to assure the highest possible degree of accuracy.

5.5.2 Flow readings for this final critical test series will be taken directly by time and weight from the calibration tank, using the Venturi Meter for secondary readings only.

5.5.3 All instruments will be calibrated per 5.2 above.

5.5.4 Data will be reduced and charted and discussed freely with the appointed personnel. Re-running of any specific point or test will be made whenever necessary.

BYRON JACKSON PUMPS, INC.

August 14, 1964"

CHAPTER 2

STUDY & SURVEY OF EXISTING INSTALLATIONS

A. PURPOSE AND SCOPE:

An investigation of high head pumping practice was conducted during the Summer and Fall of 1964 by a team, consisting of O. Hartmann, Motor-Columbus, Ltd., Baden, Switzerland and members of the Daniel, Mann, Johnson, & Mendenhall Tehachapi Project staff.

The investigation was made in order to provide more detailed data from various high head pump installations on operating and design experience, particularly with regard to reliability factors, maintenance practices, experience with various construction materials and other pertinent data.

A total in excess of thirty plants in Europe and eleven American plants were visited, most of them being plants containing pumps, although some water turbine plants having exceptionally high heads per stage and operating with Francis runners were also included. It was felt that experience with wear of runners and seal rings, etc., for the latter could be included in the overall evaluation. Particular attention was paid during the plant visits to the available suction conditions for each pumping unit. Where possible, noise measurements and vibration measurements were obtained, both during normal operation and during shut-down and start-up of the units.

See Volume No. III for detailed reports on pumping stations visited.

B. SUMMARY OF EUROPEAN PRACTICE:

1. Discussion of Purpose of Plants & Operating Records

Table 2-1 shows a list of the plants visited, including pertinent operating conditions, with the listing arranged in order of decreasing total head. Placing the plants in approximate categories, all figures referring to individual pumps, the following gives an

indication of the complexity of the installations:

a. Capacities varied from 16.5 to 1161 cfs.

b. Pumping head varied from 200 ft. to 3151 ft.

c. Number of stages in pumps varied from 1 to 9.

d. Pump power rating varied from 6800 HP to 93,400 HP.

e. Operating speeds varied from 214 to 1500 rpm.

f. Specific speed varied from 1120 to 2550.

g. Suction specific speeds varied from 834 to 8460.

The majority of the plants visited are used in pump storage service. Therefore, the pumps inspected, with one or two exceptions, are operated intermittently, and, in the majority of cases are started and stopped almost every day. Some of the units have had as many as 1000 starts and stops per year and have operated in this manner over a number of years. The plants visited have an enviable record of reliability, with unscheduled outages being practically non-existent.

This type of intermittent operation does provide sufficient time for making scheduled overhauls of the units, and this practice is the rule at all of the plants. The timing varies from once a year for minor repair to as much as ten-year intervals for plants having ideal water conditions and liberal suction submergence.

It must be understood that any comparative analysis between the European pumping practice and that of the Tehachapi Pumping Project must be considered in the light of several limitations.

All pumping stations in Europe of any magnitude are pump storage installations designed for the specific purpose of raising large volumes of water to considerably higher elevations in a relatively short time, and then shut down. Service factors, that is operating hours over totally elapsed time, are quite low, especially when compared with the expected service factor of the Tehachapi Project. (See Table 2-II)

Of the 8750 hours in a normal year, it is reasonable to expect that any pumping unit in water supply service would operate 8000 hours or something over 90% of the time. On this basis, and reviewing 75 pumping units in 27 plants which had any reasonable service factor, it was found that eight (8) pumping units in three plants had operated 60,000 hours or over, equivalent to seven years of operation. The average was 76,409 hours.

Of these eight units, four had been installed 31 years, three - 34 years and one - 39 years. During all these years the operating time varied from 26% to 31% of what might be considered a normal operating time of 8000 hours per year.

Twelve units in six plants had operated in excess of 24,000 hours, which might be considered a normal three-year period. These units, from 13 to 24 years old, had operated an average of 29,233 hours, representing 20.2% of an 8000 hour year.

Twenty units in eight plants with more than 8000 hours of pumping time had been installed an average of 8.3 years, but had operated an average of only 12,393 hours, or 18.7% of normal operating time.

Practically without exception, all plants in the Alpine area operate seasonally which allows several months each year for any maintenance or repairs that may be needed, an operation quite distinct from a water supply system, such as the MWD system or the Tehachapi system in which the service factor may be 90% or higher and a high percentage of availability must be maintained.

Time required for maintenance and inspection is rarely a problem of importance in Europe, while this problem in the Tehachapi Project will be of prime importance and must be given considerable weight.

2. Comparison of Plants with Tehachapi Project

Operating experience is available for units having approximately the same power rating as that required for the single-lift Tehachapi Project, but with the following limitations:

a. Pumps are in operation for heads higher than required for Tehachapi but for smaller capacity.

b. Pumps are in operation for capacities higher than required for Tehachapi, but operating at lower head.

The Lunersee plant is the best example of a European high head pumping plant that approximates the performance requirements of the Tehachapi single-lift system. It contains five vertical, five-stage pumping units that have a horsepower rating of 58, 400. They have a capacity of 144 CFS and operate against a head of 3151 ft. The rotational speed is 750 rpm.

The five pumps have an excellent operating record. They were placed in operation in 1957-1958 and, up to October 1964, each pumping unit had a total of approximately 14, 000 hours of operation. They have never been totally dismantled, major necessary repair work being confined to the inlet portion of the pumps which are accessible from the lower end of the pumping unit.

Two of the units furnished by one manufacturer experienced rather serious cavitation and the suction impeller of each unit was replaced by a new one after 8000 hours of operation. The replaced impellers were repaired at the plant and are available as spares.

One unit furnished by another manufacturer was repaired by welding the first stage impeller after 2300 hours of operation, without dismantling the unit. Since this first repair, the impeller had only to be polished after 12, 000 hours of operation.

No rework has been necessary on the impellers of the two other units furnished by a third manufacturer. All these units have a relatively low specific speed of 1500. The cavitation originally experienced again points out that low specific speed does not provide insurance against cavitation damage, and does not necessarily increase reliability. Cavitation is controlled by proper hydraulic design, by utilizing cavitation resistant materials, and above all, by providing adequate submergence.

There should be no problem with providing sufficient submergence for the Tehachapi Pumping Plant, and if this item is properly taken care of there is no reason whatsoever why a well designed multi-stage pump cannot give years of satisfactory service.

The trouble free operation of the Lunersee pumps (excepting some initial cavitation experienced and commented on above) points out the excellent reliability of this type of unit.

Some investigators of the Tehachapi lift have questioned whether the Lunersee pump experience is applicable because these units have a specific speed of 1500 while the Tehachapi units have a specific speed of approximately 2000. Here again it must be pointed out that specific speed is a designer's tool for designing efficient pumping units and has nothing whatever to do with reliability, etc. The Lunersee pumps operate at a speed of 750 rpm and have a horsepower rating of over 58,000, while the Tehachapi pumps for the single-lift will only operate at 600 rpm for a comparable horsepower of about 75,000. Also, the head per stage of the Lunersee pumps is 630 feet as compared to 488 ft. for Tehachapi. It may be said, therefore, that the Tehachapi pumps would be of a more conservative design than the Lunersee units.

It is conceded that a multi-stage pump does not have accessibility equal to that of a single-stage pump. However, the lack of accessibility is academic because plant experience indicates that the pumps will operate satisfactorily for many years before they must be opened for inspection and repair.

The balancing labyrinth necessary for a multi-stage pump has been of some concern as this labyrinth must withstand the full discharge pressure, and may be subject to more rapid wear than the other parts in the pump. Experience at Lunersee indicates no accelerated wear of this part, the balancing leakage having increased only 29% in 14,000 hours of operation. As normally a 100% increase in balancing leakage is considered acceptable before replacement is required, these parts should not require repairs before five years of continuous operation have elapsed. The reason for the slow wear of these parts, in spite of the fact that they are exposed to a large pressure difference, lies in the design of a long multiple labyrinth path which reduces the average velocity of the water to a very low value, thus eliminating rapid wear.

The Lunersee installation, containing five units of three different manufacturers is, according to the above, an excellent example of a

multi-stage high head installation similar to the single-lift Tehachapi application and the experience at this plant should prove without a doubt that pumps of this type are trouble-free and have a good operating record. The engineers at the station and the operating owner have confirmed this impression in writing.

There are several plants in operation with capacity-head combinations similar to that proposed for the two-lift concept, namely, Vianden, Limberg, Rodund, and Ffestiniog. These units have a good operating record; however, only one of these units, at Rodund, has operated in excess of one normal year of operation. Rodund had operated some 14, 225 hours (equivalent to 1-3/4 years of normal operation in its 12 years of life). The other 15 pumps in the three plants mentioned have an average operating life of only 3084 hours (max. 7400 hours).

Rodund operates at a 17% higher head than the Tehachapi two-lift system, but at only 60% of the capacity. This unit has a good operating record.

The Ffestiniog pumps more nearly meet the Tehachapi two-lift system conditions, but with only some 4500 hours of operation now, it will be several years before their service can be compared with what we may consider two or three years of operation at Tehachapi. Some cavitation has been experienced at Ffestiniog.

Table 2-III shows a list of the plants visited containing two-stage double-flow pumps.

There are no large single-stage units operating in Europe comparable to the projected three-lift concept for the Tehachapi Pumping Facility from which comparisons could be made, and for this arrangement American practice and experience will be evaluated in paragraph C.

TABLE 2 - I

LIST OF PUMPING PLANTS WITH PRINICPAL OPERATING DATA

(Listed in order of decreasing head)

Plant No.*	Name	Head Ft.	Cap cfs	Speed rpm	Specific Speed Ns
5	Lunersee	3151	144	750	1500
11	Tremogio	2953	16.5	1000	1120
17	Motec	2065	115	750	1270
22	Ponale	1903	130	500	1180
25	Tierfehd	1755	97	1000	1750
15 I	Z'Mutt	1541	194	1500	2140
7	Ferrera	1529	141	750	1295
24 I	Etzel	1475	92	500	1428
24 II	Etzel	1475	113	500	1581
21	Villa Gargnano	1380	487	600	1479
12	Limberg	1349	474	500	1230
14	Grimsel	1310	141	1000	1375
9	Peccia	1230	83.7	1000	1568
15 II	Z'Mutt	1200	113	1500	1980
4	Rodund	1140	353	500	1200
26 I	Sipplingen	1035	150	998	1430
26 II	Sipplingen	1035	75	1490	1532
29	Arolla	1017	148	1500	2550
23	Ffestiniog	1000	745	428	1650
20	Vianden	879	803	428	1898
32	Herva	868	357	500	1480
2	Witznau	838	353	333	1430
30	Ferpecle	700	99	1500	2340
16	Stafel	684	116	1500	1805
1	Hausern	689	353	333	1640
28	Provvidenza	565	790	500	2010
3	Waldshut	541	353	250	1490
19	Herdecke	508	494	300	1564
27	Cotilia	492	495	375	1210
31	Geesthacht	250	1161	214	1935
13	Moll	200	282	495	2410

*Reference number, signifying order that plants were inspected.

TABLE 2 - II
OPERATING HOURS

	8 Yrs. @ 8,000 Hours/per Yr.			- 64,000 Hours	
Plant	Unit	Hours	Equiv. Yrs.	Years Installed	Operational Factor
Herdecke	I	83,500 ±	10.2)		
	II	83,613	10.5)	34 yrs.	31%
	III	83,384	10.4)		
Hausern	A1	77,434	9.7)		
	A2	70,513	8.8)		
	B1	59,368	7.4)	31 Yrs.	29%
	B2	71,464	8.9)		
Tremorgio	-	82,000	10.2)	39 Yrs.	26%
Total -	611,276 Avg. 76,409			9.5 Yrs.	Avg. 28.7%

	3 Yrs. @ 8,000 Hours/per Yr.			- 24,000 Hours.	
Plant	Unit	Hours	Equiv. Yrs.	Yrs. Inst.	Oper'l.. Factor
Witznau	A3	39,936	4.9)		
	A4	36,307	4.5)		
	B3	26,869	3.4)	18 Yrs.	22.4%
	B4	27,248	3 4)		
Waldshut	A5	27,333	3.3)		
	A6	28,360	3.5)		
	B5	24,583	3.1)	13 Yrs.	17%
	B6	(23,819)	3.0)		
Etzel	IV	29,202	3.7)	17 Yrs.	22%
Provvidenza	I	24,543	3.1)	15 Yrs.	20%
Ponale	-	38,476	4.8)	24 Yrs.	20%
Herdecke	IV	24,000	3.0)	15 Yrs.	20%
Total -	350,676 Avg, 29,233			3.65 Yrs.	Avg. 20.2%

TABLE 2 - II (cont)

Plant	1 Yr. @ 8,000 Hours/per Yr. Unit	Hours	Equiv. Yrs.	- 8,000 Hours Years Installed	Operational Factor
Rodund	-	14, 225	1.78)	8 yrs.	22%
Lunersee	I	13, 500	1.69)		
	II	13, 500	1.69)		
	III	13, 500	1.69)	6 Yrs.	28%
	IV	13, 500	1.69)		
	V	13, 500	1.69)		
Grimsel	-	21, 000	2.62)	9 Yrs.	29%
Etzel	III	13, 317	1.66)	17 Yrs.	0.98%
Sipplingen	I	9, 129	1.14)		
	II	9, 777	1.22)		
	III	9, 648	1..21)	6 Yrs.	20%
	IV	9, 991	1.24)		
	V	9, 869	1.23)		
	VI	8, 771	1.10)		
Cotilia	I	17, 000	2.12)	18 Yrs.	10.7%
	II	14, 000	1.75)		
Provvidenza	II	12, 820	1.41)	11 Yrs.	12.8%
Geesthacht	I	10, 020	1.25)		
	II	9, 620	1.16)	6 Yrs.	21.3%
	III	11, 180	1.38)		
Total -	247, 867 Avg.	12, 393		1.55 Yrs.	

TABLE 2 - II (cont)

Less than one year of 8000 Hours/per Yr.

Plant	Unit	Hours	Equiv. Yrs.	Years Installed	Operational Factor
Ferrera	I	2,777	0.33)	2 Yrs.	14%
	II	1,866	0.23)		
Peccia	I	6,640	0.83)	9 Yrs.	11.8%
	II	6,580	0.82)		
Limberg	I	7,260	0.90)	7 Yrs.	13%
	II	7,400	0.92)		
Stafel	I	3,093	0.49)		
	II	4,700	0.58)	3 Yrs.	20%
	III	5,772	0.72)		
Motec	-	7,923	0.98)	5 Yrs.	19.5%
Vianden	I	3,616	0.45)	2 Yrs.	40%
	II	4,570	0.37)		
"	III	3,255	0.40)		
	IV	2,605	0.32)	1 Yr.	25%
	V	1,308	0.16)		
	VI	771	0.10)		
"	VII	151	0.02)		
	VIII	1,145	0.14)	>1 Yr.	X
	IX	753	0.09)		
Villa Gargnano	I	640	0.08)		
	II	1,170	0.14)	>1 Yr.	X
Ffestiniog	I	4,427	0.55	3 Yrs.	18.3%
	II	3,880	0.48	2 Yrs.	24%
	III	2,342	0.30	1 Yr.	30%
	IV	2,784	0.35	2 Yr.	17.5%

TABLE 2 - II (cont)

Less than one year of 8000 hours/per Yr.

Plant	Unit	Hours	Equiv. Yrs.	Years Installed	Operational Factor
Tierfehd	I	700	0.09)	>1 Yr.	
	II	700	0.09)		
Ferpecle	I	1,722	0.21)		
	II	1,401	0.18)	>1 Yr.	
	III	1,730	0.22)		
Arolla	IA	2,788	0.35)		
	IB	1,216	0.15)		
	II	2,127	0.27)	1 Yr.	25%
	III	1,627	0.20)		
Herva	-	2,000	0.25)	2 Yrs.	12.5%
Total.	103,437	Avg. 2,955		0.37 Yrs.	

NONE

Z'Mutt	I
	II
	III
	IV

TABLE 2 - III
DOUBLE FLOW TWO STAGE PUMPS
(Listed in order of Decreasing Capacity)

Plant No.	Name	Surface or Under-ground	Horizontal or Vertical	CFS	H.	HP	RPM
20	Vianden	U	H	803	880	9 x 92,800	428.6
23	Ffestiniog	S	V	745	1000	4 x 93,600	428
28	Provvidenza	U	H	565	790	2 x 58,000	1500
19	Herdecke	S	H	495	508	4 x 33,000	300
21	Villa Gargnano	U	V	487	1400	2 x 85,000	600
12	Limberg	S	H	474	1349	2 x 82,900	500
32	Herva	U	H	357	870	1 x 39,500	500
4	Rodund	S	H	353	1140	1 x 53,600	500
15	Z'Mutt	U	V	194	1541	2 x 37,200	1500
29	Arolla	S	H	148	870	3 x 19,000	1500
14	Grimsel	U	V	141	1310	1 x 25,000	1000
15	Z'Mutt	U	V	113	1200	2 x 17,000	1500

14 Pumps - 1000' to 1550'
15 " - 800' to 900'
4 " - 508'

3. Surface versus Underground Plants

Almost one-half of the plants visited were underground installations. In the majority of the cases this arrangement was dictated by the local terrain, no space being available for locating a surface plant without extensive site preparation work. Safety from enemy attack was also mentioned as an additional reason for underground installations. In general, the underground plants are well arranged and sufficient space is available for dismantling of the units for repairs.

4 Horizontal versus Vertical Shaft Arrangement

Of the twenty-eight pumping plants included in this report, six contain vertical units while twenty-two have pumps with a horizontal shaft arrangement. The latter includes Vianden where nine horizontal units with a pump rating of 93, 400 HP each are installed in an underground station.

Where general conditions favor a horizontal shaft installation, that is where the pump inlet pressure conditions are generous and where topography is favorable, this type of plant is much preferred by the operators. With all of the major operating equipment and controls located on one floor level, the supervision and maintenance work is obviously simplified. The maintenance hours for overhualing and the outage times for this work are considerably reduced.

It is also noted that the horizontal type units of the two-stage, double flow pumps on which inspections were made, seem to have a better operating record than the vertical units of this type.

5. Operating Speed and Specific Speed of Units

Plant I shows the speed of the various units investigated, the speed varying from 250 rpm to 1500 rpm, depending on the size of the units and on their age. The chart is arranged to show the unit having the lowest specific speed on the left and the one with the highest specific speed on the right.

In general, the higher specific speed units are ones installed more recently, although it must be realized that the operating speed on a

storage pump installation, which includes the majority of the units visited, is not selected for the optimum pump operation but may be picked to favor the value best suited for turbine operation.

The use of specific speed in the range of 2,000 for optimum pump design, which was selected for the basic Tehachapi program is, therefore, in line with current European practice.

6. <u>Noise Level of Installations</u>

Wherever possible, noise level readings were obtained during the plant visits. At most installations the operators were very cooperative in placing the pumping units on the line for us and in many cases, it was also possible to make noise measurements during the changeover from turbine operation. The instrument used was a portable General Radio Sound-Survey Meter, Type 1555A.

The noise reading was obtained at a distance of approximately 10 feet from the pump in all cases. The "C" scale of the meter, which gives the total sound pressure level in decibels varied from a minimum of 92 at Peccia to a maximum of 108 - 110 at Villa Gargnano, and 105 at Ffestiniog. Other values are given on the plant data pages in Volume III, and on Plate V at the end of this chapter.

It should be realized that an increase of 3 decibels represents a doubling of the noise level and any level above 96-98 decibels is considered objectionable by the average operator.

It is extremely difficult to establish the exact source of the noise on a pumping unit. Some of the observed noise levels may have been caused by the valving and header arrangements rather than the pump proper.

It may be said in conclusion that the noise levels observed during the European inspection trip were relatively high and several installations would be considered objectionable from a noise standpoint for a U. S. installation.

7. Vibration Record of Installations

Vibration readings were taken at the majority of the plants visited both during normal pumping operation and during the changeover from pump to turbine operation where applicable. The instrument used was a Davey Hand Vibrograph, Serial 941, with which a permanent record of amplitude and frequency of vibration is obtained on a small tape.

The principal amplitudes of the vibration obtained is shown on Plate V. In general, the record of most units in this regard is extremely good. Under normal pump operation, the total amplitude of vibration was, in most cases, less than 0.001 inches and in many cases was less than 0.0002 inches which is about the limit of sensitivity of the instrument. During starting and changeover procedures, the amplitude of vibration increased to a maximum of approximately .002" measured at the bearing supports, which is still well within the acceptable maximum of .002 inches to .003 inches for machinery in the speed range of 1500 rpm and below. Records of the actual vibration readings obtained are included in the reports of individual plants in Volume III.

8. Starting

The vast majority of the European installations, 19 out of a total of 27, were arranged for starting by the main turbine on the same shaft, or by an auxiliary turbine.

Of the nine plants under regular operation started electrically, four were of units under 10,000 HP, and none had units of over 30,000 HP. So they are hardly comparable to the 70,000 - 80,000 units proposed for the Tehachapi two-lift or single-lift systems. Further, of these five plants with units of over 10,000 HP, only one, that of Grimsel, had operated more than 5700 hours, (8 months of full time operation). Grimsel (29,000 HP) during nine years of service had operated some 21,000 hours, a 30% service factor, but had been started only about 40 times per year for a total of approximately 360 times. No electrical troubles were reported.

Two relatively large plants, those of Cotilia 40,000 HP and Provvidenza No. III, 67,000 HP, were originally arranged for direct reduced voltage starting. Cotilia was started once electrically and then the reduced voltage equipment was dismantled and, thereafter the units were started by the turbines.

The 67,000 HP unit at Provvidenza, having a reversible pump turbine, was at first started with reduced voltage electrically. However, after approximately 928 hours of operation and 200 starts, the amortisseur windings expanded from heat and the use of this unit as a pump was discontinued. Arrangements are now being made to use synchronous starting on this unit.

9. Quality of Water Handled by European Pumps

The pumping plants inspected were located in various parts of Europe, and handle water of many types containing varying amounts of solids.

In general, the plants do not keep detailed records of the water handled. Water quality in Germany and Austria is generally good with small amounts of solids. In the Alps of Switzerland there exists a surprising difference in water quality for seemingly similar overall conditions Although the pumps at Ferrera and Peccia handle water containing glacial silt, they do not experience excessive wear at these locations. In the Canton of Wallis, the problem of wear seems to intensify. Samples of water were obtained from the forebays of Grimsel-Oberaar and Stafel and the report following Paragraph 14, by Truesdail Laboratories, shows an analysis of these waters. The report also shows an analysis of the silt in the Plant of Fionnay which is a Francis turbine installation with a rated head of 1492 feet. The microscopic analysis of the water shows the varying nature of the silt, and points out that it has a hardness in excess of that of glass, which explains the wear on the impellers and other parts experienced at these plants.

10. Cavitation and Wear

Cavitation in the inlet of the impellers to the first stage has been experienced in a great many of the installations. This could not, in general, be observed first hand, and the reports of the plant operators have to be used as a basis for analysis in most cases.

Plate I shows which plants have reported cavitation.

First, it should be realized that the word "cavitation" is used rather indiscriminately by many of the engineers contacted. True cavitation can only occur at such points in the flow path where the absolute pressure is below the vapor pressure of the liquid handled. Impeller

blade pitting can, however, also be observed in cases where the vapor pressure is not reached, but where the water handled is corrosive. The corrosive action is accelerated at the lower pressures.

It would seem that at several plants where cavitation was reported, the cause of the impeller pitting was actually corrosion, particularly in view of the fact that changing the impeller material to stainless steel corrected the condition in these cases. The chart indicates these plants had "wear and corrosion" rather than cavitation.

At Grimsel-Oberaar a different condition exists. Photographs taken by Sulzer Bros. of an impeller and a diffuser after 20,000 hours of operation clearly show erosion at the blade tips and, in view of the nature of the water at this plant as reported in the Truesdail Laboratory report after paragraph 14, this is not surprising. By courtesy of Kraftwerk Oberhasli, these photographs are shown in the Grimsel-Oberaar section in Volume III.

This discussion should again bring to the foreground the extreme importance of the wear test program being conducted by Daniel, Mann, Johnson, & Mendenhall. Unless the approximate nature of the water handled by a pump is known, no reliable recommendation on the best design and material combination for continuous trouble free operation can be made.

11. Cavitation & Inlet Conditions

An attempt was made during this investigation to find a correlation between available suction pressure and cavitation damage.

Plate I shows the suction specific speed calculated for the various installations from available plant data on suction submergence.

It would seem from this preliminary investigation that a suction specific speed of not more than 7000 should be adopted and this will mean a suction submergence for the Tehachapi pumps of approximately 10 feet more than proposed previously by the various pump manufacturers. Of course, the final recommendation must also take into account the results of the model tests. The specifications for the model tests call for a visual inspection window for observing the early stages of cavitation, a precaution that was not taken on many of the earlier model tests conducted both in Europe and the United States, where cavitation was later experienced with the prototype installation.

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[1] Plates found at the end of this chapter.

The investigation does not show any clear trend of intensified cavitation problems with increase in specific speed Rather, it points out the extreme importance of having sufficient submergence on the selected pumps for the Tehachapi Pumping Plant regardless of the specific speed finally selected.

12. Assembly & Construction Details

a. General

Wherever possible the construction details of the pumps at the various plants were studied by inspecting the detail drawings. Some of the more pertinent details are discussed in the report.

b. Installation of Pumps

The question of horizontal versus vertical installation has already been discussed in Section 4. At some installations the pump casings were embedded in concrete but for the majority of the plants this was not the case, and no difference could be detected in vibration between the two methods. Accessibility for maintenance and repair is, of course, much improved for an open installation. It was noted that general European practice was to have two cranes available for each machine hall to facilitate handling of the parts during assembly procedures.

c. Packing Arrangement

A large number of the pumps inspected are equipped with carbon ring seals for the packing. In general, these seals have been a source of some difficulty and require periodic maintenance, once a year or so. In many cases, filtered water from special sources is required to increase the service life.

In later years, the trend indicated a change from carbon rings to labyrinth type of packing of varying designs, from single serrated bushings to multiple bushings, either smooth or serrated. One manufactuerer has used mechanical seals successfully on a large installation since 1962.

It should be noted that soft packing, which is in standard use for large United States pump installations, is not used in Europe.

d. Impeller Wearing Rings

The impeller wearing rings, as can be expected, are also a source of difficulty, although to a much lesser extent than the shaft packing. Where 12% stainless steel was used for these parts, the maintenance has been surprisingly low, even at those installations that handle abrasive water. For instance, at Grimsel-Oberaar the clearance in the rings increased from 1 m/m to 1.5 m/m in 10 years or 20,000 hours of operation.

The design of the wearing ring labyrinths vary greatly between manufacturers, some using interlocking labyrinths and others using stepped diameters for increased sealing effect. The Daniel, Mann, Johnson, & Mendenhall model test program, which includes models of various manufacturers, should contribute to an evaluation of the different designs.

13. Maintenance

The question of maintenance has already been touched upon in Sections 1, 3, 5, and 9.

The main reasons for pump maintenance in descending order of importance are:

a. Packing

b. Impeller wear or cavitation.

c. Impeller wearing rings

d. Other miscellaneous items like mechanical failures, bearing failures, etc.

Due to the nature of operation of the pumping plants visited, maintenance can be scheduled without affecting the plant availability and the availability factor has been in excess of 95% in most cases.

14. Summary of Data

Pertinent information which was collected is shown on Plates III, IV and V of this report. A chart showing in graphical form the comparison of various pertinent parameters is shown on Plate I of this report.

[1] Plates found at the end of this chapter.

REPORT

TRUESDAIL LABORATORIES, INC.

CHEMISTS - BACTERIOLOGISTS - ENGINEERS

4101 N. FIGUEROA STREE
LOS ANGELES 6
CAPITOL 5-4148

CLIENT Daniel, Mann, Johnson & Mendenhall
3325 Wilshire Blvd.
Los Angeles, California 90005 Attn: Mr. Ray D. Bowerman

DATE October 8, 1964

RECEIVED September 30, 1964

SAMPLE Water sample - Stafel - July 22, 1964
Water sample - Grimsel Lake - July 21, 1964
P.O. No. 12396

LABORATORY NO. 68501

INVESTIGATION Analyses as shown.

RESULTS

Water Analyses:

	Stafel	Grimsel Lake
pH	7.59	7.60
Specific Conductance, micromhos at 25°C	53.2	224
Total Solids, parts per million, calculated	31.9	134
Total Hardness, parts per million	32.2	32.6
Total Alkalinity as $CaCO_3$, parts per million	15.0	4.0
Corrosion Index	-1.55	-2.03

(Continued)

Silt Analyses:

(1) Stafel

Microscopic analysis of this silt indicated that it consisted chiefly of colorless crystals with the general appearance of white sand. The major portion of the particles was in the 50 to 150 micron size. The silt was insoluble in mineral acids, and showed a hardness somewhat greater than glass. This was determined by placing some of the material between two microscope slides, and with moderate pressure, rubbing the two slides together. Slight scratching of the slides was observed.

The major mineral constituents were not determined, since the hardness, particle size, etc. would appear to be the more significant criteria of the abrasive action of the silt. The crystals were definitely sharp and well defined, and would be expected to be moderately abrasive to softer materials.

(2) Grimsel

Microscopic analysis of this silt indicated the same general appearance of the Stafel silt except that the particles were very much smaller. The majority of crystals were in the particle size range of 10 to 25 microns. They were slightly harder than glass, insoluble in mineral acids, and would not be expected to be strongly abrasive to softer materials. Their fineness would, however, permit penetration into small interstices, such as bearing surfaces in machine parts, and abrasion could then result.

(3) Fionnay

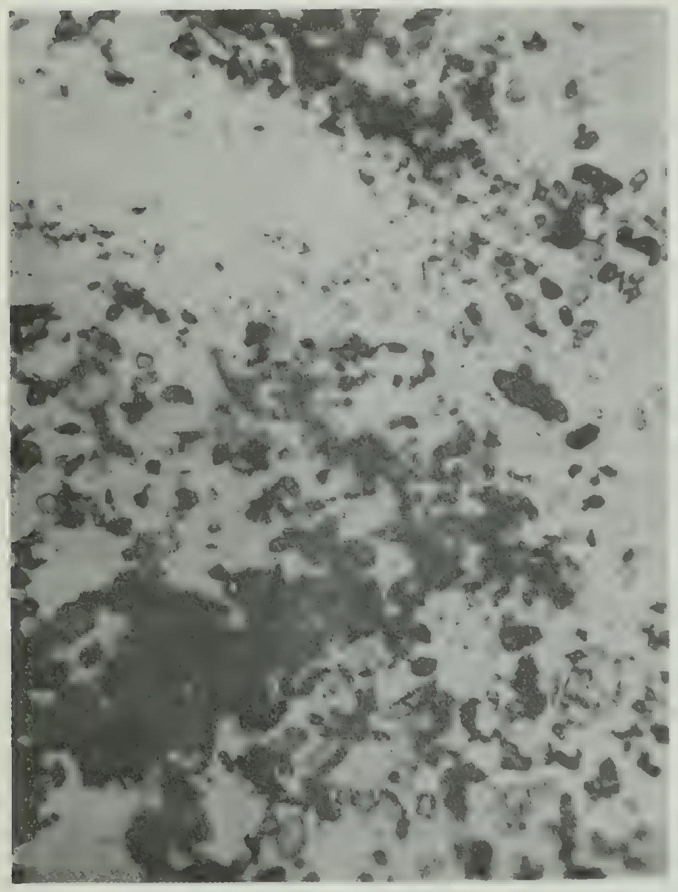
This silt appeared to be chiefly colorless sharp-edged crystals mixed with mica-like fragments and dark siliceous materials. About 10-20% was soluble in hydrochloric acid, with effervesence, indicating some limestone particles.

The major portion was in the particle size range of 150 to 1000 microns. It was considerably more abrasive to glass than the Stafel and Grimsel silts.

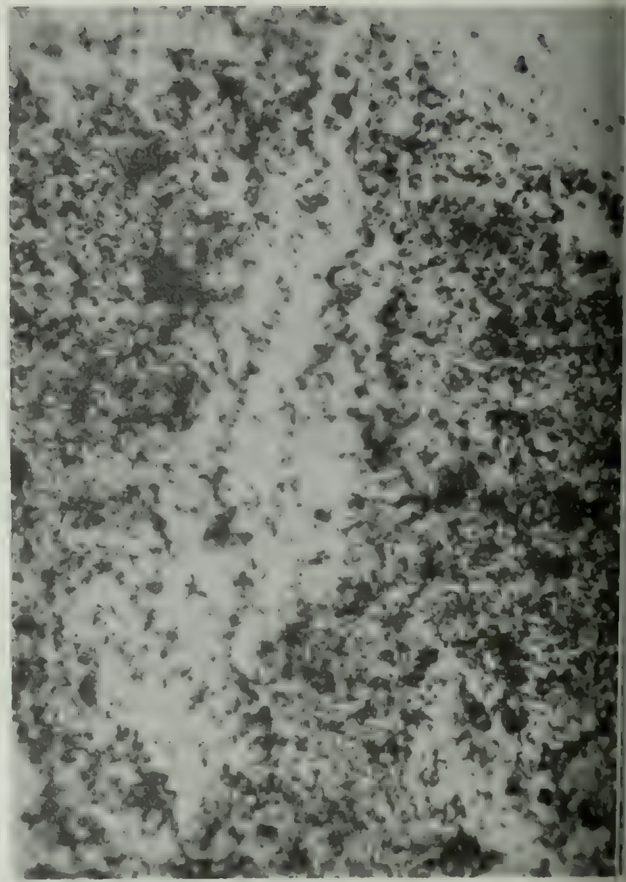
This silt would be more erosive, especially from impact, than the other two silts, but would be less apt to penetrate bearing surfaces.

Specimens of the silt were photographed with transmitted light at a magnification of 70X. The same magnification was used on all samples so that direct comparisons of the particle sizes could be made. Poor detail is shown in the Grimsel silt because of the fineness of the particles. The photographs are shown on following page.

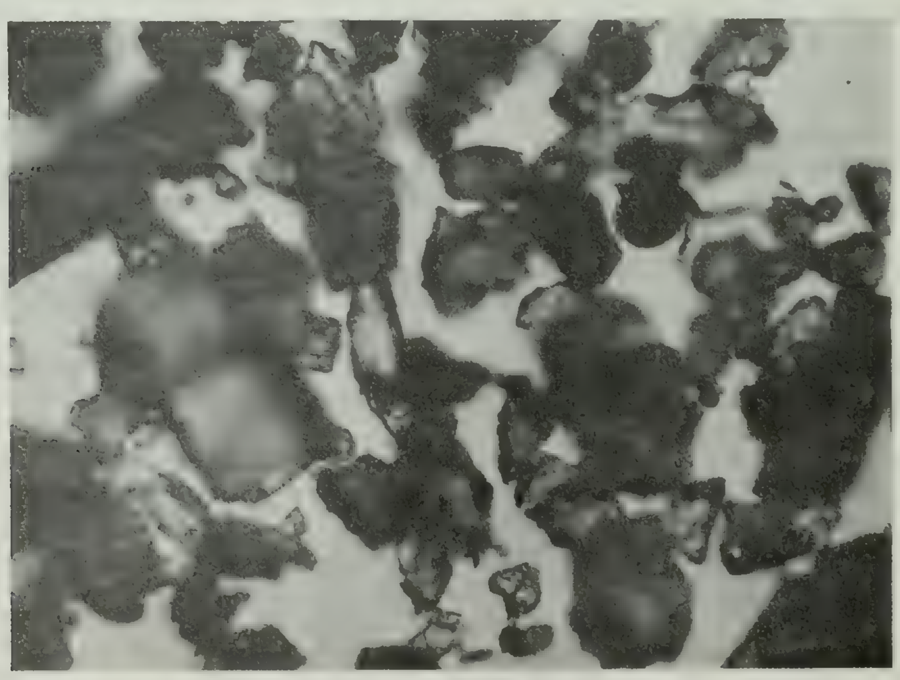
(Continued)



Stafel 70X



Grimsel 70X



Fionnay 70X

TRUESDAIL LABORATORIES, INC.

Daniel, Mann, Johnson & Mendenhall Laboratory No. 68501

REMARKS: In our opinion, the two waters are not highly mineralized, but their negative Corrosion Index values indicates that they are corrosive in nature.

The erosive qualities of the silts were discussed in the context of the report.

Respectfully submitted,

TRUESDAIL LABORATORIES, INC.

Philip J. Charley, Ph. D.
Vice President

C. SUMMARY OF AMERICAN PRACTICE:

1. Discussion of Purpose of Plant & Operating Records

Table 2-IV shows a list of the plants visited including pertinent operating conditions.

It was recognized that no station in America would closely correspond with the Tehachapi Crossing Project, but as there are points of similarity in all large pumping stations it was felt advisable to glean all the information possible from the American stations.

A total of eleven (11) pumping stations, containing 73 pumps, were visited and the operating personnel interviewed, particularly with regards to reliability factors, maintenance practices, experiences with various construction materials and other pertinent data.

The plants visited varied in heads of from 85 feet to 810 feet, in capacities from 200 CFS to 3900 CFS and in power requirements from 4300 HP to 240, 000 HP. Rotative speeds varied from 105. 9 to 450 rpm. All pumps were vertical single suction, single stage.

Four of the plants visited, Lewiston, Hiwassee, Buchanan Dam and Taum Sauk were pump-storage plants, and although of relatively high horsepower rating and high capacity, were of little value from an experience standpoint as they had little or no operating time, less than 5000 hours maximum.

Five of the stations observed were those operated by the Metropolitan Water District in the Colorado River Aqueduct. These 45 pumps have from 35, 000 to 100, 000 hours of operation with a high service factor and, therefore, their operating experience is fairly valuable. However, the maximum rating is only 12, 500 HP and the maximum head and capacity are much below those for the Tehachapi job.

The Grand Coulee Pumps present an example of a station of high power capacity and a fairly extensive time of operation, but the head is relatively low in comparison to any of the Tehachapi concepts. The Tracy pumps have operated many hours but, here again, the head is low.

None of the American pumping plants are what might be termed "underground" stations.

All plants visited, except those of The Metropolitan Water District, are operated as pumps either seasonally or periodically. The Metropolitan Water District keeps one unit per station (out of nine) down all the time for repair and maintenance.

TABLE 2-IV

AMERICAN PUMPING STATIONS

No.	Name	No. Pumps	CFS	H	HP	Operating Hours
				300	9,000	30,000-
1A-5A	Colorado Aqueduct	45	200	146	4,300	100,000
				440	12,500	
6A	Lewiston	12	3400	85	37,500	4,062
7A	Hiwassee	1	3900	205	102,000	Few
8A	Tracy	6	850	197	22,500	35,000
9A	Grand Coulee	6	1350	311	65,000	20,000
10A	Buchanan Dam	1	835	120	13,450	4,500
11A	Taum Sauk	2	2450	810	250,000	Few

2. Water Quality

Generally speaking, the quality of the water handled by the American pumping stations is fairly good to excellent. All pump storage plants pump to and from rather large reservoirs where most of the solids have settled out. This is also true with the straight pumping plant at Grand Coulee. The Metropolitan Water District water starts out clear and free from abrasive materials, but may at times accumulate a little sand enroute through the desert.

3. Cavitation & Wear

The Grand Coulee installation shows some cavitation damage which was corrected by an overlay of stainless steel, after 6000 hours of operation.

The Metropolitan Water District experienced some wear on the original bronze impellers, but none on the later pumps fitted with stainless steel impellers which have now operated some 45,000 hours.

The Tracy pumps were fitted with manganese bronze impellers and show very little wear after 13 years of operation (35,000 hours).

The American experience parallels that of Europe, in that wear is minimal when perfectly clear water is pumped regardless of the materials used, and that stainless steel wearing parts resist both wear and a certain degree of cavitation even though the water may contain some abrasive material.

4. Starting

Where the power supply system can withstand the jolt, most American pumps are started across-the-line, usually at full voltage. This includes the relatively small Metropolitan Water District units (12,500 HP maximum), the twelve 37,500 HP units at Lewiston, the 102,000 HP unit at Hiwassee, and the six 22,500 HP units at Tracy. In all cases, where the units have been started a sufficient number of times to draw any conclusions, some trouble has been experienced with heating, and consequent expansion and contraction of elements of the amortisseur windings.

It must be taken into consideration that all American pumping units of any reasonable size are of relatively slow rotative speed, 106 to 200 rpm and, therefore, they present a somewhat simpler problem as far as starting is concerned than the higher speed units proposed for the Tehachapi project.

The six 65,000 HP units at Grand Coulee and the 13,450 HP Buchanan Dam unit are started by the synchronous method, in which the pumping unit is locked together electrically with a turbine unit at standstill (or at a low speed) and the two units are brought up to operating speed together.

The 102,000 HP, 105.9 rpm Hiwassee unit is arranged for across-the-line starting with reduced voltage and with the pump de-watered. However, this unit has never operated as a pump as yet, except just a few hours during the test.

The 250,000 HP Taum Sauk units are started unwatered by direct connected induction motors of the wound rotor type.

In general, the survey of American pumping plants, and European ones too, for that matter, did not provide a yardstick to evaluate the starting problems likely to be encountered in the 70,000-80,000 HP 600 rpm units proposed for the single or double lift Tehachapi project.

5. Summary of Data

Pertinent information which was collected is shown on Plates III, IV and V of this report. A chart showing in graphical form the comparison of various pertinent parameters is shown on Plate II of this report.

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[1] Plates found at the end of this chapter.

D. SUMMARY & CONCLUSIONS PERTAINING TO EUROPEAN & AMERICAN PRACTICE:

1. Comparison with Tehachapi Plant

There are no plants in existence which are similar to the Tehachapi single-lift application. The units inspected that approach this most closely are the five vertical, five-stage units at Lunersee. While these units have only operated a total of approximately 14,000 hours each to date, they have been in operation for six years and have been giving very satisfactory performance. There should be no question that, with present manufacturing know-how, the multi-stage unit required for the Tehachapi single-lift concept could be built and would give satisfactory and reliable service for years to come.

The pumps tentatively selected for the two-lift systems, namely, the two-stage, double flow pump, have a history of very good operation in Europe and, here again, there is no question that a satisfactory and reliable unit of this type can be procured for the Tehachapi concept.

The single-stage pump required for the three-lift concept has a good precedent in American plants, and there is no doubt that this type of unit should give equally satisfactory service for the 650 ft. head selected for this arrangement. While this head is somewhat higher than that of the exisiting installations, it should be in a range where extrapolation of previous experience can be accepted.

The picture changes when the single-stage pump is considered for the two-lift system which will require a head of 975 feet in one stage. Installations of this type having a head in excess of 500 feet are few, with none of them having extensive operation experience and all of them are reversible pump turbine units.

Five high head test plants and their basic performance data are listed on the following page.

Plant	Owner	No. Units	CFS	Head	Speed	N_s
Taum Sauk	U. E. Co.	2	2450	810	200	1400
Yards Creek	JCP&L	3	2100	710	240	1700
Cornwall	Cons. Ed. Co.	8	2550	1020	257	1520
Cabin Creek	P. S. of Col.	2	1375	1058	360	1520
Oroville	DWR	3	1870	592	189.5	1450

Of these, only Taum Sauk is in operation and very little operating experience is available from this plant.

It would, therefore, seem to be unwise to select a unit of this type for the Tehachapi Pumping Plant where raw water must be pumped continuously. No precedent having a good operating record is available for this concept, and it is extremely likely that excessive wear would be encountered on the impellers and the wearing rings of this type of unit. The results of the wear test program being conducted at the Tracy Fish Intake, when completed, should throw some further light on this subject.

2. Operating Speed and Specific Speed of Unit

The specific speeds now selected for all three concepts of the Tehachapi program are around 2000. Specific speed is discussed in more detail in Chapter 3B (Volume II). As can be seen from Plates I and II of this Section, the plants visited both in Europe and the United States of America have specific speeds ranging from 1200 to as high as 4950. Generally, the units installed more recently have a higher specific speed than the ones installed earlier, although there are exceptions. However, no relation can be found between reliable operation and specific speed and it therefore seems logical to select a specific speed in the range of 2000 in order to obtain maximum efficiency for Tehachapi. This selection is consistent with the latest up-to-date practice in both Europe and the United States.

3. Cavitation and Inlet Conditions

Cavitation damage has been observed at a great many of the installations both in Europe and in the United States. The model testing for most of these installations was not conducted with visual inspection, and many of the units evidently did not have enough submergence to eliminate cavitation. On the first model test conducted under this program at Voith, definite cavitation could be observed on the impeller inlet before any evidence of noise or effect on the characteristic curve could be observed. It will, therefore, be extremely important to evaluate the cavitation test data for all three models and establish the minimum submergence required for Tehachapi, so that cavitation problems on these units may be eliminated.

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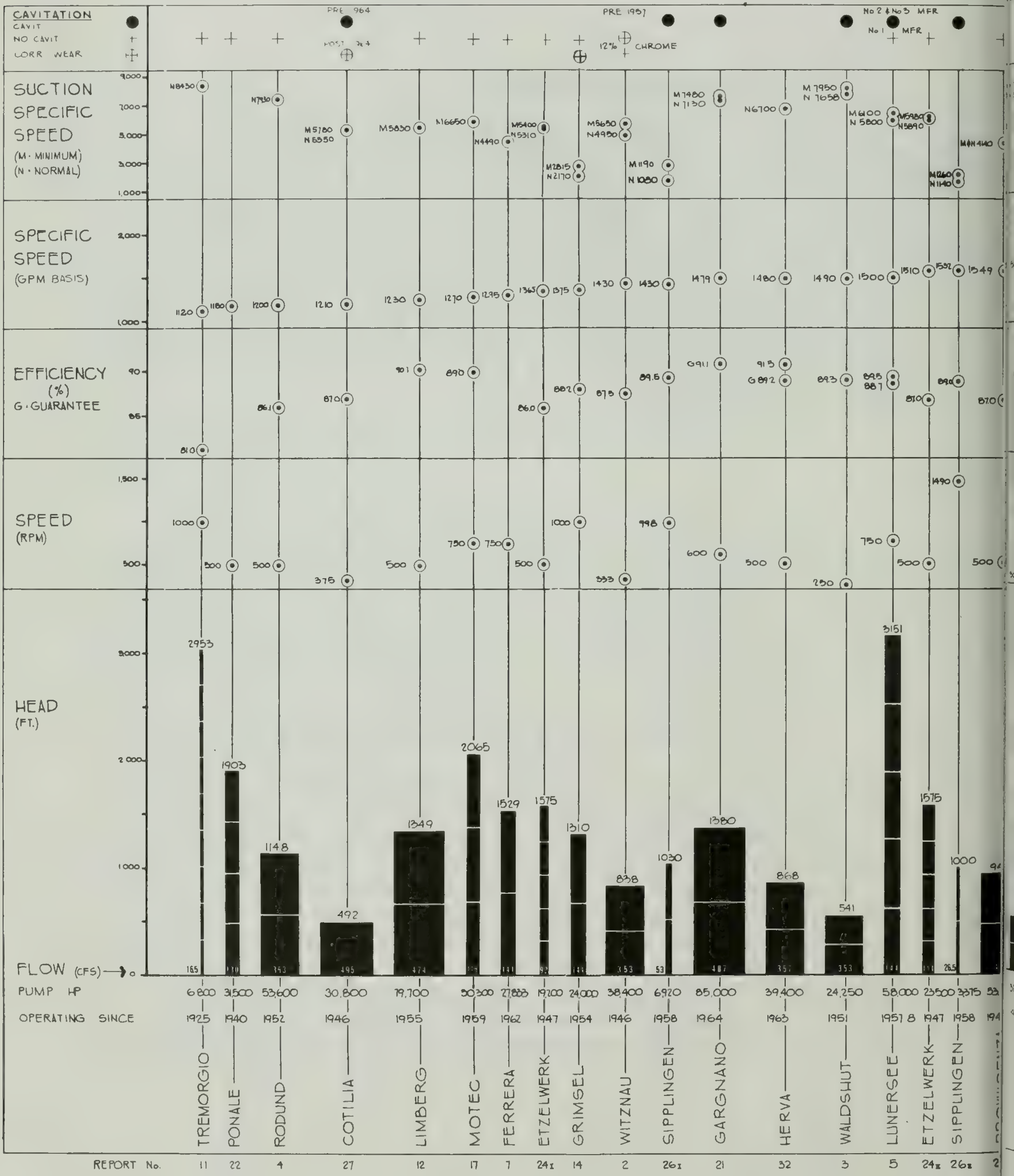
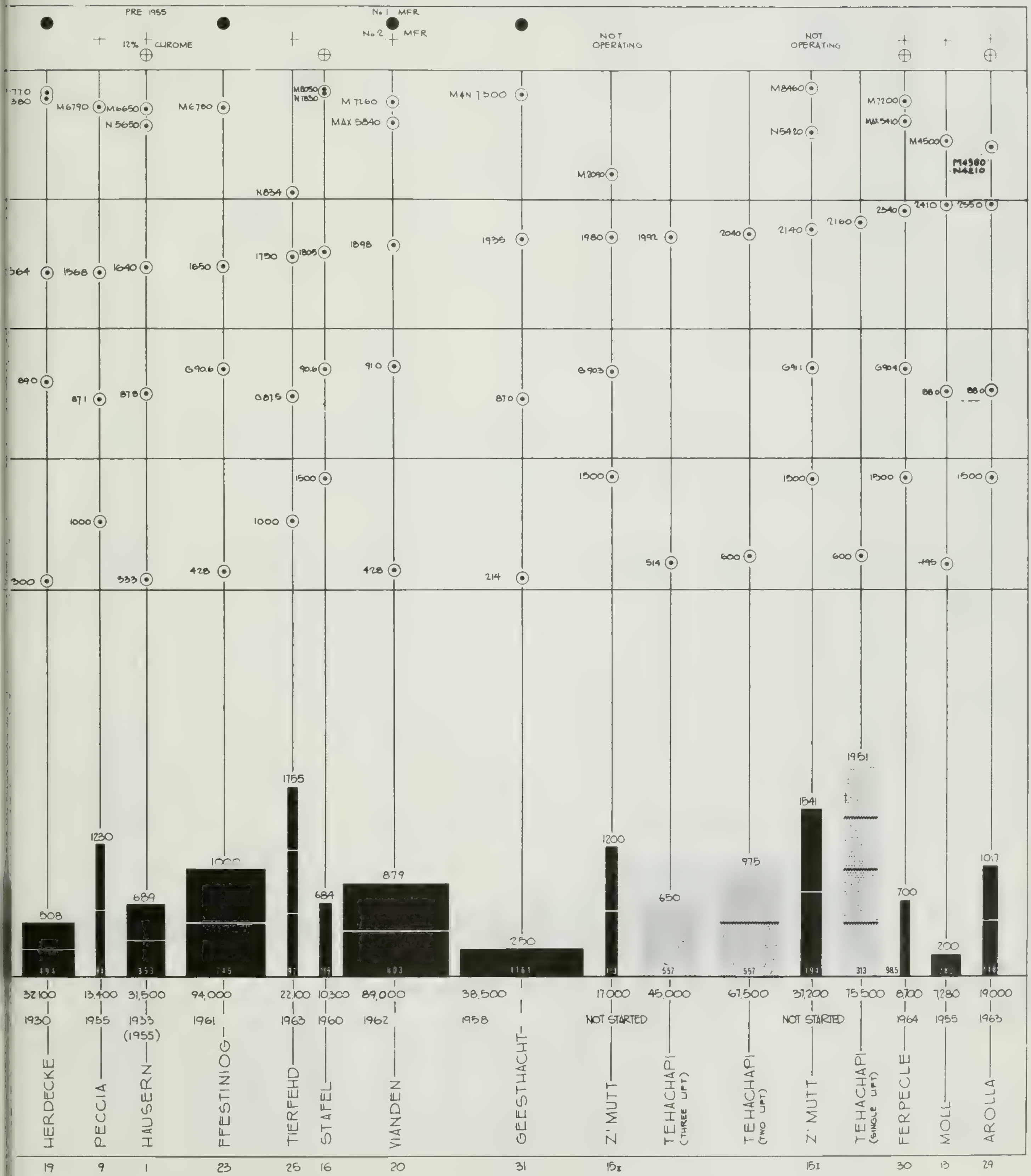
TEHACHAPI PUMPING PLANT

COMPARISON WITH INSTALLATIONS IN EUROPE




HERDECKE
PECCIA
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TIERFEHD
STAFEL
VIANDEN
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Z'MUTT
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TEHACHAPI (SINGLE LIFT)
FERPECLE
MOLL
AROLLA
PRE 1955
12% CHROME
No 1 MFR
No 2 MFR
NOT OPERATING
NOT OPERATING
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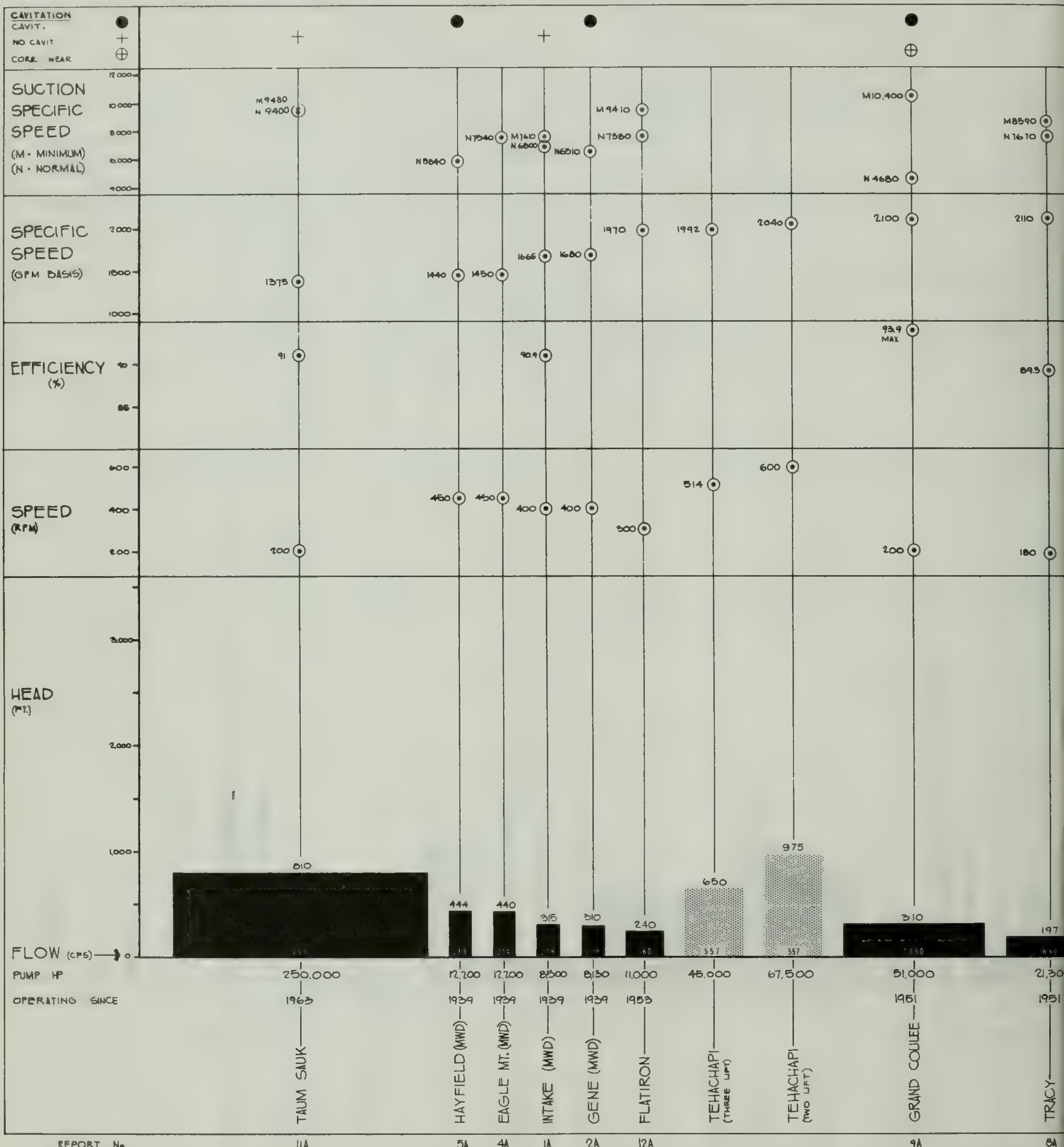
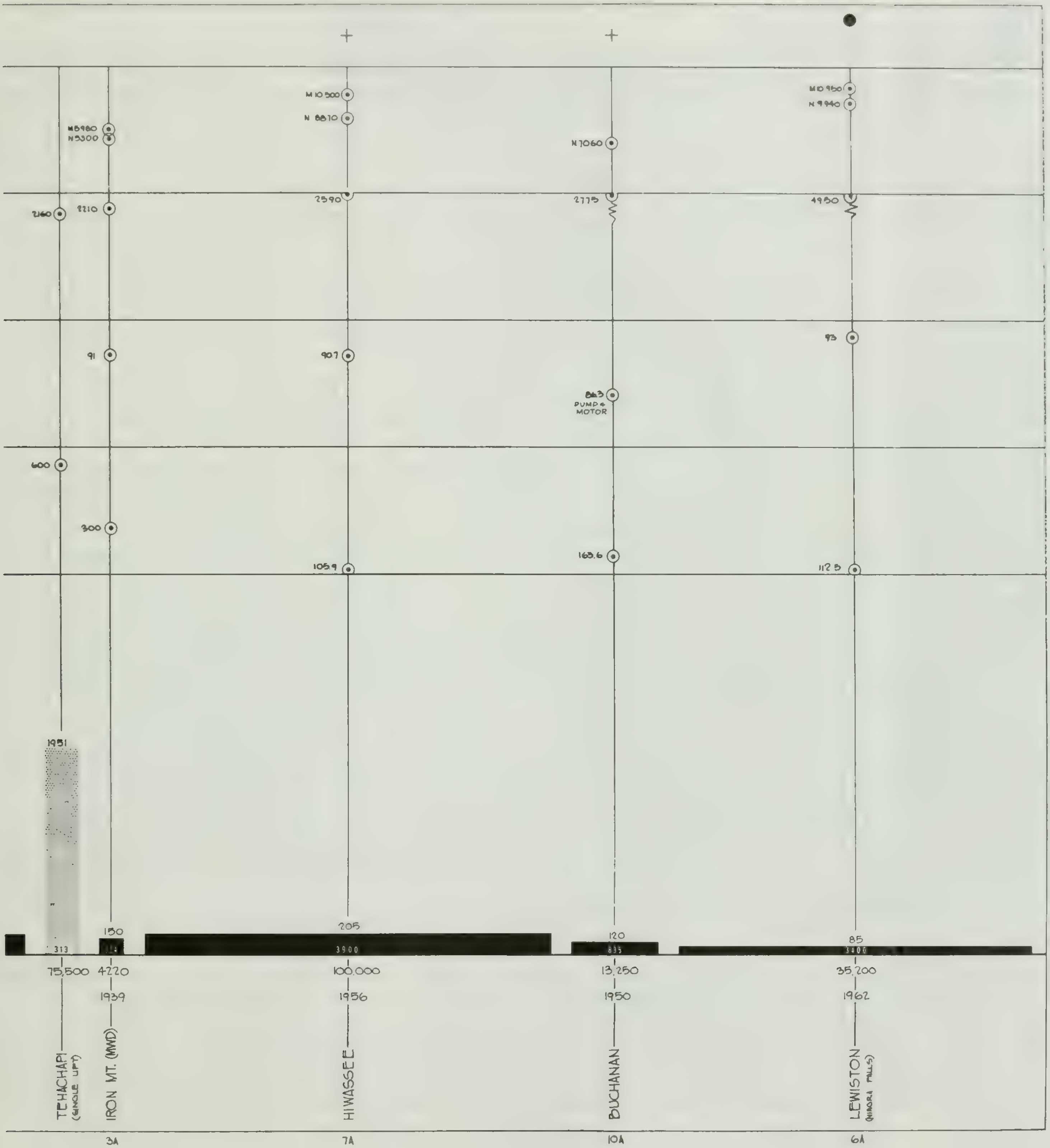
TEHACHAPI PUMPING PLANT

COMPARISON WITH INSTALLATIONS IN THE UNITED STATES

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4220
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13,250
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1939
1956
1950
1962
TEHACHAPI
(SINGLE LIFT)
IRON MT. (MWD)
HIWASSEE
BUCHANAN
LEWISTON
(NIAGRA FALLS)
3A
7A
10A
6A

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TEHACHAPI PUMP RESEARCH STUDY

SUMMARY OF DATA FOR EUROPEAN AND AMERICAN PUMPS

REPORT NUMBER	NAME OF PLANT	PLANT LOCATION (1)	OWNER AND LOCATION (1)	TYPE OF PLANT (4)	ELEVATION OF FOREBAY	YEAR OPERATION STARTED	UNITS NO	UNITS TYPE (2)	UNITS ARRANGEMENT (3)	GENERATOR OR MOTOR VOLTAGE KV	GENERATOR OR MOTOR MANUFACTURER	MANUFACTURER	UNIT POWER RATING HP	CAPACITY CFS	RATED HEAD FT
1	HAUSERN	SCHWARZWALD, G	SCHLUCHSEEWERK AG FREIBURG BREISGAU, G	S	2372	1933 REBUILT 1955 58	4	2PS	H	05	2 BBC 1 SSW	2 VOITH 2 ESCHER WYSS	31500	353	[illegible]
2	WITZNAU			S	556	1946	4	2PS	V	05	BBC	ESCHER WYSS	38400	353	838
3	WALDSHUT			S	018	95	4	2PS	H	10 5	SSW	VOITH	24,50	353	54
4	RODUND	NEAR SCHRUNS A	VORARLBERGER ILLWERKE AG BREGENZ A	S	215	1952	1	2PD	H	04	SSW	VOITH	55600	353	148
5	LUNERSEE			S	3260	1957 58	5	5PS	H	104	2 AEG 3 ELIN	2 VOITH 2 ESCHER WYSS 1 SULZER	58000	144	315
6	SILS	NEAR THUSIS S	KRAFTWERKE HINTERRHEIN AG THUSIS S	S	2 85	10/60 1/61 5/6, 9/61	4	FT	V	05	OERLIKON	ESCHER WYSS	[illegible]	64	1350
7	FERRERA	INNERFERRERA S		U	4734	1962	2	2PS	H	05		ESCHER WYSS	27833	41	529
8	BARENBURG			S	3480	1960	4	FT	V	05	BROWN BOVERI	CHARMILLES THEODORE BELL	83900	75	1110
9	PECCIA	NEAR LOCARNO, S	MAGGIA KRAFTWERKE AG LOCARNO, S	U	3390	1955	2	2PS	H	12 0	OERLIKON	SULZER	13400	83 7	1230
10	CAVERGNO			U	3393	1935	4	FT	H	3 4			45000		
11	TREMORGIO	NEAR RODI FIESSO S	CANTON OF TESSIN, S	S	6000	1925	2	9PS	H	84		ESCHER WYSS	6800	16 5	2953
12	LIMBERG	NEAR KAPRUN A	TAUERNKRAFTWERKE AG SALZBURG, A	S	5480	1955	2	2PD	H	10 25	ELIN	ESCHER-WYSS	79700	474	1349
13	MOLL			U	6680		2	1PD	H			ANDRITZ	7280	282	200
14	GRIMSEL-OBERAAR	NEAR GRIMSEL PASS, S	KRAFTWERKE OBERHASLI AG INNERTKIRCHEN S	U	6260	1954		2PD	V	3 5	BBC	SULZER	24000	141	1310
15	Z MUTT	NEAR ZERMATT, S	GRANDE DIXENCE SA LAUSANNE, S	S	I 6340 II 6770	1965	I 2 II 2	I 2PD II 2PD	I V II V		I OERLIKON II SECHERON	SULZER	I 37200 II 17000	I 94 II 13	I 154 II 1200
16	STAFEL	STAFEL S		S	7220	1961	3	1PD	H	5 0	[illegible]	SULZER	0300	116	684
17	MOTEC	NEAR CHIPPIS, S	KRAFTWERKE GOUGRA AG SIDERS, S	S	5130	1958	I 1 II 1	I 3PS II 2PS 1DW	I H II V	9 0	BBC	SULZER	I 30300 II 1270	I 51 II 1151	I 2064 II 854
18	FIONNAY	WALLIS, S	FORCES MOTRICES DU MOVOISIN SION, S	U	6440	1956	3	FT	H	10 5	BBC	ESCHER-WYSS	35300	406	1492
19	HERDECKE	NEAR ESSEN, G	RHEINISCH WESTFALISCHES ELEKTRIZITATSWERK AG ESSEN, G	S	316	1930	4	2PD	H	1 25	2 SSW 2 AEG	VOITH SULZER (JOINTLY)	32100	494	[illegible]
20	VIANDEN	NEAR VIANDEN, L	SOCIETE ELECTRIQUE DE L OUR L	U	745	1962	9	2PD	H	38	4 SSW 3 AEG 2 ACEC	5-ESCHER-WYSS 4 VOITH	69000	603	879
21	VILLA GARGNANO	WEST SHORE OF LAKE GARDA I	ENTE NATIONALE PER L ENERGIA ELETTRICA (E N E L) I	U	210	1964	2	2PD	V	100	CGBE	ESCHER WYSS	85000	487	380
22	PONALE	NORTH SHORE OF LAKE GARDA I		S	20	1940	1	4PS	H		CGBE	RIVA	3 500	30	1903
23	FFESTINIOG	WALES, E	CENTRAL ELECTRICITY GENERATING BOARD, LONDON, E	S	600	1961	4	2PD	V	16 0	AEI	SULZER	9400	745	000
24	ETZELWERK	ALTENDORF S	ETZELWERK AG ZURICH S	S	342	PUMPS 1947 TURBINE 1937	2 / 6	5PS / 1	V	100	3 BBC, 3 OERLIKON	2 SULZER ESCHER WYSS 4 ESCHER WYSS 2-BELL	I 19200 II 23500	I 92 II 113	575
25	TIERFEHD	LINTHAL, GLARUS	KRAFTWERKE LINTH-LIMMERN AG GLARUS, S	U	4250	1963	2	3PS	H	95	BROWN BOVERI, BADEN	SULZER, BROTHER WINTERTHUR	22100	97	1775
26	SIPPLINGEN	LAKE CONSTANCE, G	ZWECKVERBAND BODENSEE WASSERVERSORGUNG STUTTGART VAIHINGEN HAUPTRASSE	S	1284	1958	I 4 II 2	2PS	H	60	SSW BROWN BOVERI	VOITH	I 3375 II 6920	I 265 II 530	I 1000 II 1030
27	COTILIA	CENTRAL ITALY	ENTE NATIONALE ENERGIA ELECTRICA ROME I	U	1323	1946	2	1PD	H	10 0	I BBC II CGE	ESCHER-WYSS-TOSI	31800	495	492
28	PROVVIDENZA			U	3480	1949	2	I 2PD II PT	H V	5 0	ERCOLE MARELLI & C A	I ESCHER-WYSS II ALLIS CHALMERS	I 53500 II 65000	I 438 II 600	I 940
29	AROLLA	VAL D HERENCE, WALLIS S	GRANDE DIXENCE SA LAUSANNE S	S	5700	1963	I 2 II 2	I 2PS II 2PD	H	7 0	SECHERON	I SULZER II SULZER	19000	50	1000
30	FERPECLE			U	2800	1964	3	1PS	H	5 0	BBC	ESCHER-WYSS	8700	98 5	700
31	GEESTHACHT	22 MILES WEST OF HAMBURG G	HAMBURG ELEKTRICITATSWERK AG HAMBURG, G	S	2	1958	3	PD	H	0 0	SIEMANS SCHUCKERT	ESCHER-WYSS	38500	116	250
32	HERVA	EAST END OF SOGNE FJORD, N	ARDAL & SUNDAL VERK SA OSLO, N	U	3339	1963	1	2PD	H	8 0	AEG	RIVA	39400	357	[illegible]
1A	INTAKE	LAKE HAVASU CALIFORNIA	METROPOLITAN WATER DISTRICT OF SOUTHERN CALIFORNIA LOS ANGELES, CALIFORNIA	S	446	I 1939 II 1958 III 1956	I 6 II 2 III 1	PS	V	69	5 GE 3 ELLIOT	BYRON JACKSON	I 7550 II 8500 III 8100	I 200 II 215 III 210	294 315 315
2A	GENE	NEAR PARKER DAM CALIFORNIA		S	735	1939	I 6 II 3	PS	V	69	I GE II ELLIOT	BYRON JACKSON	7750 8130	I 200 II 215	310
3A	IRON MT	70 MILES WEST OF PARKER DAM CALIFORNIA		S	897	1939	I 6 II 3	1PS	V	69	ALLIS CHALMERS	ALLIS CHALMERS	I 3750 II 4220	I 200 II 224	146 150
4A	EAGLE MT	10 MILES WEST OF LAKE HAVASU CALIFORNIA		S	963	1939	I 4 II 2 III 3	1PS	V	69	5 WESTINGHOUSE 3 ELLIOT	WORTHINGTON	I 11300 II 12200 III 12650	I 200 II 214 III 220	440 440 447
5A	HAYFIELD	3 MILES WEST OF DESERT CENTER CALIFORNIA		S	328	1939	9	1PS	V	69	5 WESTINGHOUSE 4 ELLIOT	WORTHINGTON	12,200	215	444
6A	LEWISTON (NIAGRA FALLS)	ONE MILE EAST OF NIAGRA RIVER NEW YORK	POWER AUTHORITY OF THE STATE OF NEW YORK	S	550	1962	12	PS	V	32	6 ALLIS CHALMERS 6 S MORGAN SMITH	ALLIS CHALMERS	35,200	3400	85
7A	HIWASSEE	WESTERN NORTH CAROLINA	TENNESSEE VALLEY AUTHORITY EDNEY BLDG CHATTANOOGA TENN	S	1271	1956	1	1PS	V	13 8	ALLIS CHALMERS	ALLIS CHALMERS	100,000	3900	205
8A	TRACY	TRACY CALIFORNIA	US BUREAU OF RECLAMATION SACRAMENTO CALIFORNIA	S	4	1951	6	1PS	V	3 6	ALLIS CHALMERS	WORTHINGTON	21,300	850	197
9A	GRAND COULEE	86 MILES WEST OF SPOKANE WASHINGTON	US BUREAU OF RECLAMATION EPHRATA WASHINGTON	S	280	1951	6	PS	V	38	4 WESTINGHOUSE 2 GE	BYRON JACKSON	5,000	1350	311
10A	BUCHANAN	CENTRAL TEXAS	LOWER COLORADO RIVER AUTHORITY P O BOX 1153 AUSTIN, TEXAS	S	[illegible]	1950		PS	V	6 9	WESTINGHOUSE	WORTHINGTON	13,250	835	120
11A	TAUM SAUK	SOUTH OF ST LOUIS MISSOURI	UNION ELECTRIC CO ST LOUIS MISSOURI	S	767	1963	2	PS	V	38	GE	ALLIS CHALMERS	250,000	2450	810
12A	FLATIRON	10 MILES WEST OF LOVELAND COLORADO	US BUREAU OF RECLAMATION, LOVELAND COLORADO	S	5461	1954		PT	V		ALLIS CHALMERS	ALLIS CHALMERS	[illegible]	360	240

(1) COUNTRY
- A AUSTRIA
- E ENGLAND
- G GERMANY
- I ITALY
- L LUXEMBOURG
- N NORWAY
- S SWITZERLAND

(2) SYMBOLS FOR UNIT TYPE
- P PUMP
- T TURBINE
- S SINGLE FLOW
- D DOUBLE FLOW
- NO BEFORE LETTER INDICATES NO OF STAGES
- EXAMPLE: 2PD IS A 2-STAGE-DOUBLE FLOW PUMP
- F FRANCIS TURBINE
- I IMPULSE TURBINE (PELTON)
- DW DEEP WELL PUMP

(3) TYPE OF INSTALLATION
- V INDICATES VERTICAL SHAFT
- H INDICATES HORIZONTAL SHAFT

(4) TYPE OF PLANT
- S SURFACE
- U UNDERGROUND

PLATE III

HEAD AT START FT	RATED SPEED RPM	SPECIFIC SPEED	SUCTION HEAD NORMAL	SUCTION HEAD MINIMUM	NPSH FT NORMAL	NPSH FT MINIMUM	SUCTION SPECIFIC SPEED NORMAL	SUCTION SPECIFIC SPEED MINIMUM	SIZE DISCHARGE INCHES	EFFICIENCY MODEL GUARANTEED	EFFICIENCY MODEL ACTUAL	EFFICIENCY PROTOTYPE GUARANTEED	EFFICIENCY PROTOTYPE ACTUAL	MOTOR DIAMETER	MOTOR RATING	FLOW MEASUREMENT	INTAKE ARRANGEMENT	REMARKS
[illegible]	[illegible]	64	[illegible]	244	[illegible]	[illegible]	[illegible]	6650	[illegible]		88		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
[illegible]	[illegible]	43	452	368	[illegible]	[illegible]	465	5650	[illegible]		83		[illegible]			[illegible]	[illegible]	
[illegible]	[illegible]	490		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]				[illegible]				[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	594	[illegible]	8950	474			846	[illegible]			[illegible]	[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]	686	3	[illegible]	580	[illegible]	25.6 MFG 31.5 MFG 2 & 3			[illegible]	[illegible]			25 CURRENT METERS	[illegible]	
[illegible]	[illegible]								[illegible]			[illegible]	[illegible]			[illegible]	[illegible]	
[illegible]	[illegible]	295	[illegible]		45		449		295									
110	[illegible]																	
[illegible]	[illegible]	[illegible]	MAX 683	590	MAX 970	[illegible]	MAX [illegible]	6790	[illegible]			[illegible]	[illegible]			[illegible]	[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]		219		8430						[illegible]					
[illegible]	[illegible]	[illegible]	MAX 3940	59	MAX [illegible]	850	MAX [illegible]	[illegible]	[illegible]			890	[illegible]			[illegible]		[illegible]
[illegible]	495	[illegible]		590		830		[illegible]					88				[illegible]	
[illegible]	[illegible]	375	[illegible]	2290	[illegible]	[illegible]	[illegible]	[illegible]	276			[illegible]	88			[illegible]		
[illegible] 60	530	24 [illegible]	[illegible]	[illegible]	[illegible]	[illegible]	592	[illegible]	[illegible]			[illegible]	[illegible]					
[illegible]	500	835	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	276	[illegible]	89	[illegible]	[illegible]					
[illegible]	[illegible]	[illegible]	495	495	[illegible]	757	[illegible]	6650	[illegible]			[illegible]	[illegible]			[illegible]	[illegible]	
492	750																	
[illegible]	[illegible]	564	[illegible]	[illegible]	[illegible]	300	7380	[illegible]	[illegible]				[illegible]				[illegible]	
4385	4246	898	MAX 666	[illegible]	MAX 963	730	MAX 5840	[illegible]	7.0			[illegible]	[illegible]			[illegible]	[illegible]	
690	600	[illegible]	[illegible]	[illegible]	840	[illegible]	[illegible]	[illegible]	[illegible]			[illegible]					[illegible]	
476	[illegible]	[illegible]	[illegible]															
500	420	[illegible]	MAX [illegible]	43		76		6780	66			906		[illegible]	[illegible]		[illegible]	
[illegible]	500	[illegible]	208	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]			86.0	[illegible]			[illegible]	[illegible]	[illegible]
59	1000	750	MAX [illegible]	558	602	588	MAX 820	834	2.6			88					[illegible]	
[illegible]	[illegible]	[illegible]	59	[illegible]	345	303	[illegible]	[illegible]	[illegible]				89.0 [illegible]				[illegible]	
492	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	5550	[illegible]	[illegible]		[illegible]		[illegible]			[illegible]	[illegible]	
[illegible]	[illegible]	[illegible]	98.4	98.4	265	265	4.40	[illegible]	[illegible]		86.0		87.0			[illegible]	[illegible]	
[illegible]	[illegible]	[illegible]	236	[illegible]	65.2	[illegible]	42	438	[illegible]			[illegible]	[illegible]			THERMODYNAM	[illegible]	[illegible]
[illegible]	560	[illegible]	MAX [illegible]	[illegible]	MAX 226	54	MAX 5410	[illegible]	[illegible]			90.4				THERMODYNAM	[illegible]	
[illegible]	[illegible]	[illegible]	49	49	[illegible]	[illegible]	[illegible]	7300	65	[illegible]	[illegible]	[illegible]	87.0				[illegible]	
434	[illegible]	480	59	26.2	58		[illegible]		5.2			89.2	[illegible]			[illegible]	[illegible]	
[illegible]	400	[illegible]	[illegible]		48	4.6	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]			[illegible]	[illegible]	
[illegible]	400	[illegible]	[illegible]		[illegible]		6270 6510		[illegible]		[illegible]		9.2				[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	40	[illegible]	[illegible]	[illegible]		[illegible]		9.4			[illegible]	[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]		49		[illegible]		46.5		90.2		90.0				[illegible]	
484	[illegible]	[illegible]	[illegible]		435		5840		43.5		90.2		90.7				[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]				[illegible]	
[illegible]	[illegible]	253	[illegible]		556	[illegible]	[illegible]	[illegible]	[illegible]		[illegible]	[illegible]	[illegible]		14.9	[illegible]	ONE SHORT PENSTOCK	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	8590	[illegible]			88	[illegible]				[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	888	[illegible]	44	[illegible]	[illegible]	[illegible]	[illegible]			[illegible]	[illegible]	[illegible]
[illegible]	636	[illegible]	[illegible]		[illegible]		7060		84			[illegible]	[illegible]			[illegible]	[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	63	[illegible]	[illegible]	[illegible]	[illegible]				[illegible]				[illegible]	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	94 C					[illegible]			[illegible]	[illegible]	

DMJM
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PLANNING • ARCHITECTURE • ENGINEERING • SYSTEMS

TEHACHAPI PUMP RESEARCH STUDY
SUMMARY OF DATA FOR EUROPEAN AND AMERICAN PUMPS

REPORT NO	NAME OF PLANT	MATERIALS										RADIAL CLEARANCE		IMPELLER DIAMETER INCHES	DIA OF SHAFT AT	
		CASING	IMPELLER	IMPELLER WEARING RINGS		SEAL RINGS		SHAFT PACKING		DIFFUSER	SHAFT	WEARING RINGS (INCHES)	SEAL RINGS (INCHES)		PUMP BEARING (INCHES)	IMPELLER (INCHES)
				STATIC	ROTATING	SEAL	SLEEVE	SEAL	SLEEVE							
1	HÄUSERN	CAST STEEL	1933 BRONZE STEEL 1955 STAGE 1 13% CR STAGE 2 CAST STEEL	1933 C.I 1955 C.S	1933 NONE 1955 13% CR	CAST STEEL BABBIT LINED	CAST STEEL BABBIT LINED	4 CARBON RINGS	1933 BRASS 1955 C.I		STEEL	0.039	0.008	100 EYE 63	19	20
2	WITZNAU	CAST STEEL	1846 CAST STEEL 1957 STAGE 1 13% CR 1% NICKEL	STEEL	STEEL	STEEL	STEEL	4 CARBON RINGS			CARBON STEEL	0.039	0.008	111-11 VANES 64" EYE	19	22
3	WALDSHUT	CAST STEEL	CAST STEEL STAINLESS WELD	STEEL	STEEL						STEEL	0.039	0.008			21.7
4	RODUND	CAST STEEL	13% CHROME	CAST IRON	3% CHROME	BABBIT LINED		3 CARBON RINGS	BRONZE WITH CR. PLATE	CAST STEEL	CARBON STEEL	0.039	0.08	83"-13 VANES 86"-11 VANES		
5	LÜNERSEE	CAST STEEL	13% CHROME 1% NICKEL	13% CHROME	13% CHROME	BRONZE BABBIT	BRONZE BABBIT	EW&V CARBON SULZER BABBIT	CR STEEL	13% CHROME	CARBON STEEL	0.02-0.03	0.02	51.3"-59.2"		
6	SILS *	CAST STEEL	13% CHROME	13% CHROME	13% CHROME			3 CARBON RINGS	BRONZE		STEEL	0.039	0.039	84"		21.6
7	FERRERA		13% CHROME													
8	BÄRENBURG *															
9	PECCIA	CAST IRON	13% CHROME		BRONZE	BRONZE	CAST IRON	BABBIT LABYRINTH			CARBON STEEL		0.02	49 27 EYE		11-11.8
10	CAVERGNO *															
11	TREMORGIO															
12	LIMBERG	CAST STEEL	ST I 13% CHROME ST 2 MN STEEL	MN STEEL	MN STEEL			NOW, BABBIT	ORIG BRONZE NOW CR PLATE	MN STEEL		0.035		90		22.2-23.6
13	MÖLL		CHROME STEEL													
14	GRIMSEL-OBERAAR	CAST STEEL	13% CHROME	CAST IRON LABYRINTH	13% CHROME	BRONZE	CAST IRON LABYRINTH	BABBIT LABYRINTH		13% CR STEEL		0.039	0.02	51		15.75
15	Z'MUTT		13% CHROME											STAGE 1 36.2 STAGE 2 37.5		11.8
16	STAFEL	CAST STEEL	13% CHROME	CAST STEEL	ALUMINUM BRONZE	NONE	NONE	BRONZE WITH BABBIT LABYRINTH	STAINLESS STEEL		STEEL	0.0275	0.028	34 20 EYE		11
17	MOTEC	CAST STEEL	13% CHROME	CAST IRON	BRONZE	2% NICKEL CAST IRON	LABYRINTH	4 RINGS BABBIT ON BRONZE	3% CHROME	13% CHROME		0.0295	0.024	62.25		16.5
18	FIONNAY *	CAST STEEL	13% CHROME	13% CHROME	13% CHROME			3 CARBON 1 BABBIT RINGS				0.039		61		19.5
19	HERDECKE	SUCTION C I DISCHARGE C.S	ORIG BRONZE NOW S STL	CAST IRON	ORIG BRONZE NOW S STL.	NONE	BABBIT	1930 2 CARBON 1955 BABBIT	STEEL BUSHING	BRONZE	STEEL	0.039	0.059	98.5 60.5 EYE		29.5
20	VIANDEN	CAST STEEL	ST I 13% CHROME ST. 2. C.S	BRONZE	STEEL	NONE	BABBIT	3 CARBON RINGS	BRONZE		STEEL	0.036-0.042	0.031-0.038	V 93 E W 94 EYE V 28.7-60; EW 27.4-54.75		V 25.6-30" EW 27.4"
21	VILLA GARGNANO	CAST STEEL	13% CHROME							STAINLESS STEEL				78.5		23.6
22	PONALE		ST I S. STL. ST 2-4 C S	BRONZE	BRONZE	BABBIT				CAST STEEL				73		17.7
23	FFESTINIOG	CAST STEEL	13% CHROME	ALUMINUM BRONZE	13% CHROME	13% CHROME	BABBIT	LABYRINTH		STAINLESS STEEL		0.059-0.070	0.00325-0.0039	STAGE 1 96 STAGE 2 10. EYE 75"		27
24	ETZELWERK	CAST STEEL	BRONZE	CAST IRON	BRONZE	BRONZE	BRONZE	BABBIT LABYRINTH	BRONZE	CAST IRON	STEEL			64		15.75
25	TIERFEHD	CAST STEEL	13% CHROME	CAST IRON	BRONZE	13% CHROME	BRONZE	LABYRINTH GLAND		STAINLESS STEEL	STEEL			47	13.8 COMBINED WITH THRUST BRG	15.75
26	SIPPLINGEN	I C I II C S	CHROMIUM-STEEL	CAST IRON	NONE	BABBIT	BABBIT	TEFLON ASBESTOS	STAINLESS STEEL	BRONZE	STEEL	0.012		I II 25.8		I II 6.7
27	COTILIA	CAST STEEL	BRONZE	ORIG. C I NOW C S	ORIG BRONZE NOW S STL	NONE		ORIG JUTE-GRAPHITE NOW CARBON RINGS		BRONZE	STEEL	0.039		107.3 67 EYE		18.6
28	PROVVIDENZA	CAST STEEL	CAST STEEL	STAINLESS STEEL				CARBON RINGS	BRONZE					81.5 52 EYE		20.4-32.2
29	AROLLA	CAST STEEL	STAINLESS STEEL	STAINLESS STEEL	STAINLESS STEEL	13% CHROME	13% CHROME	ALUMINUM BRONZE	13% CHROME LABYRINTH	CAST STEEL	STEEL			STAGE 1 31 STAGE 2 33.4		9.9
30	FERPECLE	STEEL	13% CHROME 1% NICKEL	BRONZE	13% CHROME	BRONZE	STAINLESS STEEL	NONE	NONE	13% CHROME 1% NICKEL	STEEL	0.016-0.019	0.016-0.019	35.3	11.8	
31	GEESTHACHT	CAST STEEL (WELDED VOLUTE)	CAST MN STEEL	CAST STEEL		NONE	NONE	3 CARBON RINGS	BRONZE			0.047		141		32.9
32	HERVA	CAST STEEL (WELDED VOLUTE)	STAINLESS STEEL	BABBIT	NONE	BABBIT	NONE	5 CARBON RINGS	STAINLESS STEEL		STEEL	THEORETICALLY, 0		76		23.5
1 A	INTAKE	CAST STEEL	I BRONZE II & III S STL	ORIG C.I. BRONZE NOW CAST STEEL	ORIG. CAST IRON NOW S. STL	NONE **		Pb FOIL "ALL PACK"	STAINLESS STEEL	NONE	STEEL	ORIG 0.015 (0.02-0.025)		78.63 EYE 40		20
2 A	GENE	CAST STEEL	5-BRONZE 4-S STL	ORIG BRONZE NOW CAST STEEL	ORIG CAST IRON NOW S STL	NONE **		COTTON OR ASBESTOS FIBER	STAINLESS STEEL	NONE	STEEL	ORIG 0.015 (0.020-0.025)		78.63 EYE 40		20
3 A	IRON MT.	CAST STEEL	6-BRONZE 3-S. STL	BRONZE INSERTS STEEL	CAST IRON STAINLESS STEEL	NONE			STAINLESS STEEL	NONE	STEEL	ORIG 0.015 (0.020-0.025)		74.1 EYE 40.88		TAPERED 20.31
4 A	EAGLE MT.	CAST STEEL	I & II BRONZE III S STL	I & II C WITH BRONZE INSERTS III STEEL	I & II BRONZE III S. STL	NONE **		GARLOCK HEMP-LEAD	STAINLESS STEEL	NONE		0.020-0.025		82.9 EYE 34		22
5 A	HAYFIELD	CAST STEEL	BRONZE STAINLESS STEEL			NONE **			STAINLESS STEEL	NONE		0.020-0.025		82.9		
6 A	LEWISTON (NIAGRA FALLS)	FABRICATED STEEL	STEEL ASTM-A27 25	SAE-1045 (STEPPED)	SAE-1020 STEEL	NONE		ADJUSTABLE SOFT PACKING	STAINLESS STEEL	CAST STEEL		0.100		206-180 ANGLE CUT EYE 78.5	28	28.5
7 A	HIWASSEE	FABRICATED STEEL	ASTM A-27-60T GRADE 70-36 ANNEALED	SAE 1025	SAE 1045	NONE **		ADJUSTABLE GRAPHITE-TYPE 400	CHROME STEEL	STEEL	ASTM A235-52T CLASS A ANNEALED	0.100	NONE	267.6 EYE 182		40
8 A	TRACY	CAST IRON	MN BRONZE	ORIGIONAL MN BRONZE, NOW STAINLESS STEEL	ORIGIONAL: MN BRONZE, NOW STAINLESS STEEL	NONE		ADJUSTABLE GRAPHITE IMPREG "GARLOCK"	BRONZE	NONE		0.035-0.040		144 EYE 76.8		20.4
9 A	GRAND COULEE	FABRICATED STEEL	STEEL	BRONZE	BRONZE	NONE **		ADJUSTABLE TEFLON-ASBESTOS		YES	CARBON STEEL	ORIG: 0.035 NOW 0.100		167.38 EYE 89.38		28
10 A	BUCHANAN	CAST STEEL	BRONZE	BRONZE	BRONZE	NONE		8 RINGS		NONE	CARBON STEEL	0.044		120 EYE 64		20
11 A	TAUM SAUK	WELDED STEEL (T-1)	STEEL			NONE		ASBESTOS		NONE		0.035		260 ±		48
12 A	FLATIRON	WELDED STEEL		STAINLESS STEEL	BRONZE				STAINLESS STEEL		STEEL					

* TURBINE

** MATERIAL OF BALANCE RINGS SAME AS WEARING RINGS

ABBREVIATIONS

C I.--CAST IRON
C R--CHROME
C.S--CAST STEEL
MN---MANGANESE
ST.----STAGE
S. STL.--STAINLESS STEEL

PLATE IV

COUPLING TYPE	ARRANGEMENT OF SHAFTING	PUMP BRG SIZE & CONST (INCHES)	THRUST BRG SIZE & CONST (INCHES)	SUCTION VALVE TYPE	SUCTION VALVE MANUFACT	SUCTION VALVE SIZE & RATING	SUCTION VALVE OPERATION	SUCTION VALVE CONSTRUCTION	DISCHARGE VALVE TYPE	DISCHARGE VALVE MANUFACT	DISCHARGE VALVE SIZE (INCHES)	DISCHARGE VALVE CONTROL	DISCHARGE VALVE CONSTRUCTION	REMARKS
[illegible] CONVERTER		89 UPPER	36 MICHELL	FLAP CHECK			[illegible] OPEN HYDRAULIC	SQUARE X-SECTION	NEEDLE	2 VOITH 2 ESCHER WYSS	55	WATER & OIL OPERATION	CAST STEEL SEATS 3% CR	BALANCE RINGS STATIC CAST STEEL ROTATING 3% CHROME WITH 0.039 CLEARANCE, REPLACED AT 0.06 CLEARANCE
IMPULSE TURBINE		22 C I BABBIT LINED	MICHELL 400 TON RATING	SHUT OFF FLAP			LOCKED OPEN HYDRAULIC	SQUARE X SECTION	NEEDLE	VON ROLL	55		CAST STEEL SEATS 2% CR	
IMPULSE TURBINE				1 ROLLER GATE & 2 BUTTERFLY	VOITH VOITH	3'×7.4 4.9			NEEDLE	VOITH	55	OIL		
[illegible] GEAR TYPE		19" BABBIT LINED		STOP LOCKS			WITH CRANE		SPHERICAL	ESCHER WYSS	47.4	OIL WITH AIR PRESSURE ACCUMULATOR	STEEL	ONE [illegible] STATIC IMPELLER RINGS REPLACED BY 13% CR IN 1956
		23.6		GATE					NEEDLE	VOITH CHARMILLE	[illegible] 23.5 / [illegible] 3.25	OPEN OIL CLOSE WATER		
SOLID	3 BRG ARRANGEMENT	1 BRG AT TURBINE	IN GENERATOR 250 TON MICHELL						SPHERICAL	VON ROLL				
				BUTTERFLY	VON ROLL		OPEN OIL CLOSE WEIGHT		2 PLUNGER NEEDLE	VON ROLL	29.5	OPEN OIL CLOSE WATER		
	3 BRG ARRANGEMENT, 2 BRG MOTOR BRG PUMP	CAST IRON BABBIT LINED	DOUBLE ACTING KINGSBURY TYPE	NONE					NEEDLE & SPHERICAL	VON ROLL	23.6	NEEDLE OIL & WATER SPHERICAL WATER		BALANCE RINGS 10 LABYRINTH
														BALANCE RINGS REPLACED EVERY YR ROTATING C.I. STATIC BRONZE
		22.4-23.6	DOUBLE ACTING MICHELL	NONE					NEEDLE	VON ROLL	63	OPEN OIL CLOSE WATER		VALVE HAD BRONZE SEATS TROUBLE AFTER 5YRS PERFER BRONZE STEEL
RIGID		1 BABBIT LINED IN PUMP	KINGSBURY TYPE IN MOTOR	SPHERICAL	VON ROLL	27.6"	WATER		NEEDLE / SPHERICAL	CHARMILLE / VON ROLL	27.6 / 27.6	OIL		STUFFING BOX OPERATED FOR 10YRS WITH NO TROUBLE LEAKAGE INCREASED FROM 24 58 GPM
				BUTTERFLY		I 2-47.2" II 39.4" 55"			SPHERICAL		I 31.5 II 27.6			
SOLID		7×7	DOUBLE KINGSBURY TYPE	NONE			EMPTY RESERVOIR FOR REPAIRS		NEEDLE	CHARMILLE	27.6	OIL		
GEAR WITH HYDRAULIC OPERATION		2 BABBIT LINED 13.8 & 7.25	KINGSBURY TYPE	NONE					SPHERICAL	VON ROLL	27.5	WATER		BALANCING RINGS COMPOSED OF BRONZE
									SPHERICAL	VON ROLL	47.3	WATER	BRONZE	
1930 HYDRAULIC WITH FRICTION CLUTCH 6 PELTON TURB. WITH GEAR		COLLAR BRG	—	STOP LOCKS FOR REPAIRS		36" WIDE			1 NEEDLE 3 SPHERICAL	VOITH	67	OIL		
		22.4 BABBIT	KINGSBURY TYPE	GATE	ESCHER WYSS VOITH	57" 90.5×51.2 OVAL			4 NEEDLE 5 SPHERICAL	VOITH ESCHER-WYSS	71	OPEN OIL CLOSE WATER		
			MICHELL TYPE AT BOTTOM			71"					55			
		17	MICHELL TYPE 145 TON RATING			39.4"			NEEDLE	R VA	33.4			BALANCE RINGS LABYRINTH
GEAR WITH HYDRAULIC OPERATION		27×21	MICHELL	SLIDING GATE	RANSOMES & RAPIER	14.8×22'	GANTRY CRANE		SPHERICAL STRAIGHT FLOW	ENGLISH ELECTRIC	66	WATER		
FRICTION CLUTCH	MICHELL THRUST BEARING INCORPERATED INTO PUMP	12.6	38.5 O.D MICHELL	FLAPPER	ESCHER WYSS	31.5"×43.3"	HYDRAULIC	RECTANGULAR	2 SPHERICAL	ESCHER-WYSS	31.5, 25.6 29.5	I WATER II MANUAL		DISCH VALVE CONSISTS OF 31.5 VALVE WITH VENTURY HAVING 25.6 THROAT & 29.5 MANUAL VALVE BALANCE RINGS, ROTATATING BRONZE, STATIONARY C.I.
HORIZONTAL RIGID COUPLING		3.8 COMBINED WITH THRUST BRG	36 O.D	SPHERICAL	VON ROLL	23.6	WATER	TYPICAL VON ROLL	DOUBLE ACTING NEEDLE	VON ROLL	21.6	WATER & ELECTRIC		BALANCE RINGS ROTATING BRONZE, STATIC C.I.
HORIZONTAL ELASTIC COUPLING			FIXED SEGMENTS	BUTTERFLY	BOPPS & REUTHER	II 3 5'	MANUAL		NEEDLE	I BOPPS REUTHER II VOITH	I 19.7 II 27.5	I MECHANICAL II OPEN OIL CLOSE WATER		
		18.6×12 BABBIT LINED STEEL	KINGSBURY TYPE 28.8 O.D	2 ROTARY	R VA	2-67"	MANUAL & OIL PRESSURE		NEEDLE	TOSI	71	OPEN OIL CLOSE WATER		
		PILLOW BLOCK OIL UNDER PRESSURE	2 RINGS IN BEARING	ROTARY	TOSI SANGIORGIO	90.5"	OIL		2 IN SERIES SPHERICAL	R VA	67	OIL		
		7×7.9 C I BABBIT LINED	1 17" DISC O.D.	NONE					SPHERICAL	CHARMILLE	I 19.7 II 27.5	OIL		
RIGID		4" WIDE 11.8 DIA OIL PRESSURE	15.7 DISC	BUTTERFLY	ESCHER-WYSS	31.5"	MANUAL	OPENS AUTOMATICALLY IF DISCH VALVE OPENS	SPHERICAL	ESCHER-WYSS	24.5	OPEN OIL CLOSE COUNTER WEIGHT	SEAL IN ROTATING PART	
GEAR CLUTCH		15.5×9.7 OIL PRESSURE LUBRICATION	35.4 O.D KINGSBURY TYPE	GATE		7.3×6.6'	CRANE		NEEDLE	ESCHER-WYSS	106	OPEN OIL CLOSE WATER		200 GPM LEAK FROM STUFFING BOX WHEN PUMP NOT OPERATING
		19.6 OIL RING	27.5 DOUBLE ACTING KINGSBURY TYPE	GATE	KVAERNER BRUG		HYDRAULIC		NEEDLE	R VA	5.2	OPEN OIL CLOSE WATER		
SOLID COUPLING	INTERMEDIATE SHAFT	20×25 PUMP GUIDE	KINGSBURY 6 PADS 33 O.D IN MOTOR 60 TON LOAD	GATE		6'×6'	CRANE		TAPERED CONE	5-S MORGAN SMITH 4-WILLAMETTE	42×60'	OIL	ROTATING PLUG DAVIS DESIGN	
SOLID COUPLING	INTERMEDIATE SHAFT	20×25	KINGSBURY 33 O.D 6 PADS	BUTTERFLY		60"	MECHANICAL OR ELECTRIC MOTOR		TAPERED CONE	3 S MORGAN SMITH 6-WILLAMETTE	42×60	OIL	ROTATING PLUG DAVIS DESIGN	
SOLID COUPLING	INTERMEDIATE SHAFT	20×12	IN MOTOR 33 O.D	BUTTERFLY	WILLAMETTE	60"	MECHANICAL OR ELECTRIC MOTOR		TAPERED CONE	S MORGAN SMITH	48×60	OIL		
SOLID COUPLING	INTERMEDIATE SHAFT	22×20	IN MOTOR 33 O.D	BUTTERFLY	WILLAMETTE	60"	MECHANICAL OR ELECTRIC MOTOR		TAPERED CONE	3 PELTON CHAPMAN 6-PELTON WILLAMETTE	46.5×57"	OIL		PACKING LASTS 9 MONTHS
SOLID COUPLING	INTERMEDIATE SHAFT		IN MOTOR	BUTTERFLY		60"	ELECTRIC MOTOR OR MANUALLY		TAPERED CONE	PELTON WILLAMETTE	46.5×57"	OIL		PACKING LASTS 9 MONTHS
SOLID COUPLING		28" DIA 21.75" LONG BABBIT LINED	IN MOTOR	SLIDING GATE					SLIDING GATE		24'×24	GANTRY CRANE		NO BALANCE RINGS - BALANCED THROUGH HOLES IN HUB
SOLID COUPLING		40 25×30	KINGSBURY 84" DIA	STOP GATES					STOP GATES		19×26	HOISTS IN DAM	AT TOP OF PENSTOCK	
SOLID COUPLING	INTERMEDIATE SHAFT		KINGSBURY	BULKHEAD GATES		13×12' (2 PER PUMP)	GANTRY CRANE	STRUCTURAL STEEL	BUTTERFLY	NEWPORT NEWS SHIP BUILDING	108	OIL & AIR PRESSURE		BALANCE RINGS ORIGIONALLY MN BRONZE, NOW STAINLESS STEEL, PACKING REPLACED YEARLY
SOLID COUPLING		27.25×27 BABBIT (C.012 CLEARANCE)	IN MOTOR SPRING LOADED	REVERSE FLOW COASTER GATE					NONE			•		
SOLID COUPLING		20		SLIDING GATE					BUTTERFLY	S MORGAN SMITH	84"	D C MOTOR		PUMP DUPLICATE OF TRACY PUMPS EXCEPT FOR IMPELLER SIZE
SOLID COUPLING	3 BEARINGS		IN MOTOR	GATE			CRANE		SPHERICAL	ALLIS-CHALMERS	108"	OIL	STEEL	SPHERICAL VALVE JAMMED ON ORIGINAL START-UP NOW O.K.
				BUTTERFLY					BUTTERFLY		76			

TEHACHAPI PUMP RESEARCH STUDY
SUMMARY OF DATA FOR EUROPEAN AND AMERICAN PUMPS

REPORT NO.	NAME OF PLANT	OPERATION								WATER QUALITY				
		PUMP START CLOSED DISCH.	PUMP DEWATERED	RING LUBRICATION	REVERSE OPERATION	NOISE MEASUREMENT (db)		VIBRATION MEASUREMENT (IN)		ph	SOLIDS	HARDNESS	SALINITY	REMARKS
						OPERATION	START	OPERATION	START					
1	HÄUSERN	YES	NO	NO	NO	93	98							WATER QUALITY GOOD 11 ppm CO_2
2	WITZNAU		YES	YES	NO	93	100							WATER QUALITY GOOD
3	WALDSHUT		YES	YES	NO	95	105	0.0003	0.0015					WATER QUALITY GOOD
4	RODUND		YES	YES	YES			SLIGHT						CLEAR WATER
5	LÜNERSEE	YES	NO	NO	YES	98-100	110-112	LESS THAN 0.0002	0.0009					WATER QUALITY GOOD
6	SILLS *													WATER NOT ABRASIVE
7	FERRERA	YES	NO	NO	NO	95	98	0.0002	0.001-0.035					WATER USED FOR DRINKING AFTER FILTERING
8	BÄRENBURG													WATER NOT ABRASIVE
9	PECCIA	YES	NO	NO	YES	92	98	0.0005	0.0015					WATER VERY GOOD
10	CAVERGNO *													
11	TREMORGIO													
12	LIMBERG	YES	NO	NO		98	100-105	0.0002	0.0005					WATER COMPOSED OF GLACIAL MELT
13	MÖLL					95	100-105	0.0002	0.002					WATER NOT ABRASIVE
14	GRIMSEL-OBERAAR	YES	NO	NO	YES	102	110	0.0002	0.0008		GLACIAL SILT			SOLIDS RANGE FROM 1 PP TO 100 PPM
15	Z'MUTT													
16	STAFEL	YES	NO	NO	YES	94-97		0.0007			YES			GLACIAL MELT - SANDY
17	MOTEC	YES	NO	NO	NO	100	105-110	0.0004	0.0005		GLACIAL SILT			WATER POOR QUALITY. CONSISTING OF GLACIAL MELT AND RAIN
18	FIONNAY *										GLACIAL SILT			
19	HERDECKE		YES	YES	NO	98-100	105	0.0002	0.0002	VARIES	YES			POLLUTED WATER
20	VIANDEN		YES	YES	NO	98	105-115	0.0003	0.001			SOFT		WATER RELATIVE CLEAN CONTAINS CO_2
21	VILLA GARGNANO	YES	NO	NO		108-110 VOLUTE	115 VOLUTE	0.0008 VOLUTE	0.0030 VOLUTE					WATER QUALITY EXCELLENT
22	PONALE	YES	NO	NO										WATER EXTREMELY CLEAN
23	FFESTINIOG	YES	NO	NO		105	110	0.0015		8.32	21-26 ppm	SOFT		WATER CLEAN
24	ETZELWERK	YES	NO	NO	NO						NONE			
25	TIERFEHD	YES	NO	NO	NO						GLACIAL SILT		NO	
26	SIPPLINGEN	YES	NO	NO	NO	95-96.5	95	LESS THAN 0.0002	0.0003	7.8	NO			PURE, CLEAN LAKE WATER
27	COTILIA	YES	NO	NO	YES			SLIGHT			NO	HARD	NO	VERY GOOD WATER
28	PROVVIDENZA	YES	NO	NO		99-104	97-103	SLIGHT			NO	SOFT		
29	AROLLA	YES	NO		YES	95					YES			MILKY COLOR - CONTAINS FINE PARTICLES OF GRANITE, ETC
30	FERPECLE	YES	NO		NO	100-102	103				GLACIAL SILT	SOFT	NO	MILKY COLOR - CONTAINS FINE PARTICLES OF QUARTZ, GNEISS, ETC
31	GEESTHACHT	YES	NOT NORMALLY	YES	NO	QUIET		SLIGHT			FINE SLUDGE			POLLUTED WATER
32	HERVA	YES	NO	NO	NO	93.5-98	100-104.5				SAND	SOFT	NO	SAND EROSION EXPECTED
1A	INTAKE	YES	NO	NO	NO	84-87		NONE		7.96	776	332 $CaCO_3$		GOOD CLEAR LAKE WATER CORROSION INDEX +0.54 ALKALINITY: 117.5 ppm **
2A	GENE	YES	NO	NO	NO			NONE		7.96	776	332 $CaCO_3$		GOOD LAKE WATER; CORROSION INDEX +0.54; ALKALINITY 117.5 PPM **
3A	IRON MT.	YES	NO	NO	NO	84-88				7.88	776	335.6 $CaCO_3$		ALKALINITY 120 PPM ABRASIVE SAND CORROSION INDEX +0.49 **
4A	EAGLE MT.	YES	NO	NO	NO	84-91				7.98	755	333 $CaCO_3$		GOOD CLEAR WATER, CONTAINS LITTLE SAND ALKALINITY 70 PPM CORROSION INDEX +0.60 **
5A	HAYFIELD	YES	NO	NO		84-88	QUIET			7.95	767	338.4 $CaCO_3$		ALKALINITY 125 PPM CORROSION INDEX +0.60 SOME SAND **
6A	LEWISTON (NIAGRA FALLS)		YES	YES										POLLUTED LAKE WATER
7A	HIWASSEE		YES	YES	YES	85-94		NONE						CLEAN LAKE WATER
8A	TRACY	YES	NOT NORMALLY	YES		82-88				ABOUT 7	FINE SILT			DELTA WATER VARIES
9A	GRAND COULEE	NO	NO		YES			YES						CLEAR LAKE WATER CONTAINS SILT, NO SOLIDS
10A	BUCHANAN	YES	NO											GENERALLY GOOD, SOMETIMES TURBID
11A	TAUM SAUK		YES	YES	YES			YES						
12A	FLATIRON													760 ACRE FEET RESERVOIR

* TURBINE **RESULTS OF ONE SAMPLING

PLATE V

TOTAL HOURS X 1000		MAINTENANCE					IMPELLER				CASING WEAR	SHAFT SEAL WEAR	REMARKS
		OUTAGE				TOTAL HOURS							
OPERATION	STANDBY	PLANNED	UNPLANNED	CAUSE OF UNPLANNED	MAINT. HRS EACH PLAN'D		CAVITATION	WEAR	SEAL SEIZURE	RADIAL SEAL WEAR			
59.4-77.4	92.2-139.9	EVERY 10 YRS	2	1. IMPELLER BOLT 2. DIAPHRAM CASTING	9-10 WEEKS		1933 YES 1955 NO	1933 YES 1955 NO	NO	0.006-0.012 IN 10 YEARS	NO	0.012-0.020 IN 10 YEARS	FIRST STAGE IMPELLER POLISHED ONCE A YEAR
26.9-39.9	36-63				9-10 WEEKS		1946 YES 1957 NO	1946 YES 1957 NO					CAVITATION WEAR IN 1ST STAGE
23.8-29.3	32.1-52.2				8 WEEKS		YES	NO					CAVITATION WEAR IN 1ST STAGE
4.2	90.7	EVERY 2 YRS	0		8 WEEKS		NO	NO		NO			1ST STAGE GUIDE VANES HAD CAVITATION AND/OR CORROSION ON BOTTOM HALF
12	35	EVERY 7-9 YRS			3900 MAN HRS	3900 MAN HRS	YES	NO					CARBON RINGS REPLACED EVERY 1000 HRS LABYRETH PACKING REPLACED EVERY 9000 HRS
0	.6	0	0			0	NO	NO					
9-2.8	19-20	EVERY 2 YRS	1	STUFFING BOX GOT HOT	350 MAN HRS		NO	NO	NO				
							NO	NO					
6.6	68	0	2	1ST CHANGED PACKING BOX 2nd INCREASED CLEARANCE IN BABBIT BUSHING	1ST 660 MAN HRS 2nd 15 WEEKS		NO	NO					
							NO	NO					
82	263						NO	NO					
7.2-7.4	76	0	0				NO	NO			YES	YES	LABYRENTH SEAL RINGS REPLACED EVERY 2 YRS
							NO	NO					
21.0	62	IN 5 YRS			3 WEEKS	530 MAN HRS	NO	YES		0.02" IN 20,000 HRS	NO		IMPELLER TIPS WELDED WITH 13% CR STEEL
NOT OPERATING													
30-58	24.5-2.6	1 PER YR			1 WEEK		NO	YES		YES	NO		IMPELLER REGROUND EVERY 1000 HRS & REPAIRED EVERY 2000 HRS. WEAR RINGS REPLACED EVERY 2000 HRS
7.9	35.1	1 PER YR	3	OVERHEATING OF COUPLING	15 DAYS		NO	NO					
16.5-19.3	36-52	2					NO	NO					
83	302	EVERY 5 YRS					YES	NO	NO				
0.2-4.6	17.3-12.9	EVERY 3-4 YRS	0		700 MAN HRS		MFR* 1 YES MFR* 2 NO	NO	NO	ALMOST NONE	NO	NO	
0.6-1.2	5.4-6						YES	NO					
38.5	214		0				NO	NO					
2.3-4.4	12.1-18.3		0				YES	NO	NO	NO	NO		EXPERT TO REPAIR CAVITATION DAMAGE EVERY 3-5 YRS
3.4-29.2	136-120	0	0			0	NO	NO					
0.7	3.7-4.5	0				0	NO	NO	NO	NO			
8.8-10.0	39-40	EVERY 2 YRS	0		1 WEEK	2 WEEKS	YES	NO	NO	NO	NO	NO	
4-17	145-148	EVERY 2 YRS	2	DISCHARGE VALVE			PRE 1964 YES 1964 SLIGHT	YES	NO	NO		0.032 IN 8000 HRS	
0.9-24.5	21-111		0				NO				NO		PUMPS HAVE BEEN RUNNING 1-13 YRS WITH NO INSPECTION OR REPAIRS
2-2.8	2.4-4	EVERY 3 YRS		FAILURE OF WARM WATER VALVE GALLING OF THRUST BRG. WEAR OF BALANCING LABYRINTH			NO	YES	NO	0.039 IN 5 MONTHS	NO		
1.4-1.7	6.0-6.3	1 PER YR	0			0	NO	YES	YES	YES	YES		
0.0-11.2	46-47	EVERY 3 YRS	1	OIL LEAK IN SERVO METER COUPLING	5 WEEKS		YES			NO			IMPELLER PLATED WITH STAINLESS STEEL IN CAVITATION AREAS
2.0	20		0										
30-100	5-95	55	30	PACKINGS, HOT BEARINGS	270	15,000	NO	YES	NO	0.026 0.070			TWO ESTIMATES OF 20 YEAR SEAL WEAR
30-100	2-100	56	26	PACKINGS, HOT BEARINGS	270	5,000	YES	YES	NO	0.026" 0.070"			TWO ESTIMATES OF 20 YEAR SEAL WEAR
30-100	5-110	38	15	PACKINGS	30	1,000		YES	NO				OVERHAUL DELAYED DUE TO OPERATION DEMANDS
30-100	5-105	56	25	PACKINGS, HOT BEARINGS	125	7,000		YES	NO	0.020" 0.150"			TWO ESTIMATES OF 22 YEAR SEAL WEAR
30-100	5-100	46	35	PACKINGS, HOT BEARINGS	200	9,000	YES	YES	NO	0.020" 0.150"			TWO ESTIMATES OF 22 YEAR SEAL WEAR
4	8	PER YR	0 FOR PUMP		2 DAYS		YES			SLIGHT			CAVITATION REPAIRED WITH STAINLESS STEEL
35 PUMP 0.04	69.1	EVERY 2-3 YRS	0		25 MAN-DAYS		NO			ALMOST NONE			UNIT NOT OPERATED AS PUMP LONG ENOUGH TO DRAW CONCLUSIONS
35.0	92	EVERY 10 YRS	0 FOR PUMP		2 MONTHS		VERY SLIGHT			UPPER 0.034-0.080, LOWER 0.074-0.12			IMPELLER DEVELOPED CRACKS WITHIN 2 YRS, REPAIRED BY CHAIN LOCK METHOD - NOW O.K.
7.9-21.8	101-105		2-6	PACKING TROUBLE, EXCITER TROUBLE, STATION PIPING & OIL COOLERS, IMPELLER BALANCING			YES	YES		SOME			IMPELLER CAVITATION BOTH SIDES OF BLADES REPAIRED WITH S STL E 308 & E 309 WELDING EVERY 6000 HRS
4.5	20.3		0				NO						PUMP NEVER OVERHAULED
													UNIT NOT OPERATED LONG ENOUGH TO OBTAIN OPERATING EXPERIENCE

CHAPTER 3

PUMP DESIGN STUDIES

A. PURPOSE AND SCOPE

The process of selecting pumps for a pumping plant consists of the following steps:

1. Determine delivery requirements and seek suitable pump manufacturers

2. Evaluate the manufacturer's ability to supply equipment in a satisfactory manner.

3. Review precedents and operating experience of existing plants

4. Finalize pump characteristics, size, auxiliary requirements, etc., and write the pump specification. This is the basic "exterior view" of pump selection and is normally the extent of pump "research" necessary for a plant installation.

The Tehachapi lift requires pumps that must be custom designed and must have certain functional requirements that dictate extensive care in the process of selecting hardware. The pump designs must be reviewed in detail in order to assure that the prospective manufacturers are providing their most advanced designs and are giving utmost consideration to mechanical and structural reliability. In order to properly evaluate the pumps, an "internal review" of the designs is desirable.

Responsibility for the hydraulic and mechanical design of the pumps lies with the manufacturers, and it is to their own advantage to provide the best designs of which they are capable. However, evaluation of designs may be difficult due to conflicting performance requirements (as functions of the three different lift concepts) and to different opinions as to the importance of particular operating features. The purpose of DMJM's pump design studies is to develop sufficient background material to fairly evaluate the pump designs presented by the model testing firms. Studies are being made of the basic hydraulic design, mechanical features, cavitation performance and economic factors dependent on design.

Certain studies have been given particular emphasis and due to their scope, are treated as prime subjects in other chapters of this report. Included in this category are: Hydraulic Transients (Chapter 6); Reliability Study (Chapter 7); Methods of Estimating Efficiency from Model Tests (Chapter 8); Vibration Study

(Chapter 9); and, Materials Investigation Wear Testing (Chapter 10). At a later point of time in the Tehachapi Study, cavitation performance may be given an enlarged study and made a prime subject.

B. HYDRAULIC DESIGN

Hydraulic design is the responsibility of the pump companies participating in the model test program. Many of the internal design details are considered as proprietary information and have not been disclosed. As a consequence, the degree of the "internal" pump study is limited. However, certain knowledge of the impeller geometry and the hydraulic passages can be obtained from the cross-sectional drawings. Specific information on impeller and diffuser vane angles and curvature is not known. Some geometric information has been obtained by scaling from drawings and, consequently, must be of questionable accuracy.

Impellers have been given prime consideration. Information on diffusers, return channels, and volutes was too meager to warrant any analytical attempt. The analysis that has been made is subject to changes in design that may occur during the model testing. Before commenting on the results, a few explanatory remarks are in order:

1. Considerations for the Hydraulic Analysis

a. Background Material

Designers in the field of hydraulic machinery hold to their individual theories and pet choices of design parameters. Numerous variations are found in pump details — most of minor consequence. However, the refinement of pump design continues and the advantages of one pump over another, no matter how slight, may have important economic or other value. The importance of a small percentage of efficiency in the Tehachapi lift is an example. However, the evaluation of a pump design is a very difficult task as exemplified by the variety of opinions that can be found among the designers themselves.

The basic reason for the variable positions of pump experts is the fact that the mathematical models and analytical procedures that have been developed in the hydrodynamic field do not and cannot practically take into account all the factors of fluid properties, observed fluid behavior, and flow geometry. Hydrodynamics has traditionally relied on experimental efforts to provide basic understanding of fluid flow and to provide the design numbers used in engineering processes. Centrifugal pumps is a particularly difficult analytical subject because of the infinite variation of three-dimensional geometries that are possible. Superimposed on the design problem are

limitations of materials and manufacturing processes, mechanical elements, operational and maintenance considerations, and business economics. Centrifugal pump design, like many other engineering subjects, is still a combination of science and art and the percentage of art is high.

Somewhat in contrast to the statements of the previous paragraph, it can be said that some aspects of centrifugal pump design have been reduced to a common approach. This has come about through intensive effort by the pump industry and related technical agencies extending over a long period of time. Among the several textbooks on the subject, "Centrifugal and Axial Flow Pumps" by A. J. Stepanoff, is one of the more prominent. The methods and basic notation of that text have been used in this investigation of the proposed Tehachapi pump designs.

b. Definitions, Notation, Equations

FIG. 3-1 illustrates the notation for impeller diameters. FIG. 3-2 illustrates the discharge velocity triangle and notation.

List of Notation and Equations:

D_2 = overall impeller diameter (O.D.)

b_2 = breadth of impeller passage at discharge

D_1 = inlet diameter of impeller

D_{im} = mean diameter of flow passage at vane entrance(approx)

d_1 = breadth of flow passage at vane entrance (approx)

D_h = diameter of shaft

$V_i = \frac{Q}{A_i}$ = velocity of approach – impeller inlet

$C_{m_1} = \frac{Q}{A_1}$ = meridional velocity at vane entrance

$C_{m_2} = \frac{Q}{A_2}$ = meridional (radial) component of discharge velocity

Cu_2 = tangential component of discharge velocity

C_2 = absolute discharge velocity

$u_2 = \frac{1}{12} \frac{D_2}{2} \omega$ = tangential velocity of impeller at discharge

w_2 = relative velocity of water to impeller at discharge

$u_1 = \frac{1}{12}\frac{D_1}{2}\omega =$ maximum tangential velocity of vane at the inlet

$\omega = \frac{2\pi N}{60} =$ angular velocity of impeller

$A_2 = \frac{1}{144}\pi D_2 b_2 =$ discharge area of impeller

$A_i = \frac{1}{144}\frac{\pi}{4}\left(D_1^2 - D_h^2\right) =$ area of approach impeller inlet

$A_1 = \frac{1}{144}\pi D_{im} d_1 =$ normal area to flow at vane entrance (approx)

N = rotational speed – RPM

Q = flow rate

$\psi = \frac{H}{u_2^2/g}$ = head coefficient

g = gravitational constant

H = stage head

$\phi = \frac{C_{m_2}}{u_2} = \frac{Q}{A_2 u_2}$ = flow coefficient

$K_u = \frac{u_2}{\sqrt{2gH}}$ = discharge tangential velocity coefficient

Note: $\psi = \frac{1}{2 K_u^2}$

$K_{m_1} = \frac{C_{m_1}}{\sqrt{2gH}}$ = inlet meridional velocity coefficient

$K_{m_2} = \frac{C_{m_2}}{\sqrt{2gH}}$ = discharge meridional velocity coefficient

$N_s = \frac{N Q^{\frac{1}{2}}}{H^{3/4}}$ = specific speed

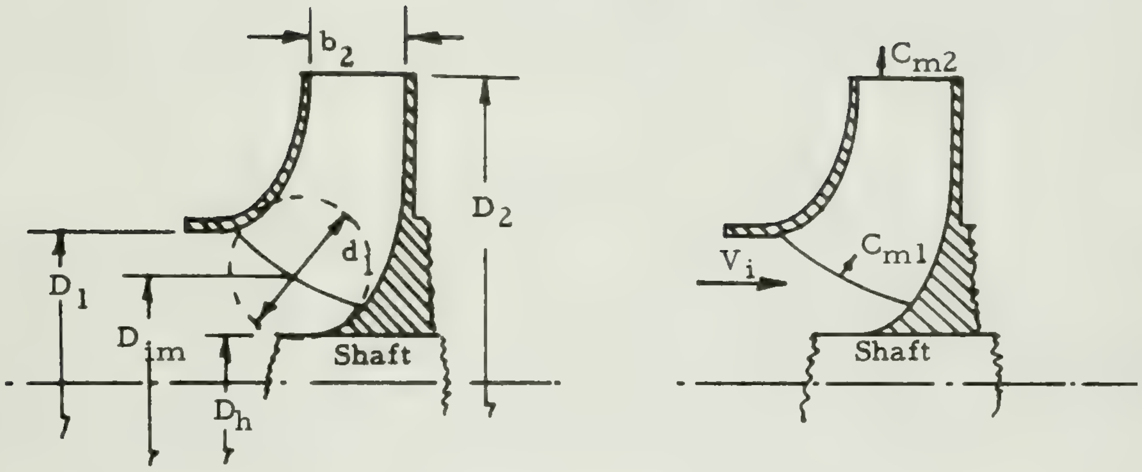


Fig. 3-1 Impeller Geometry and Water Velocities

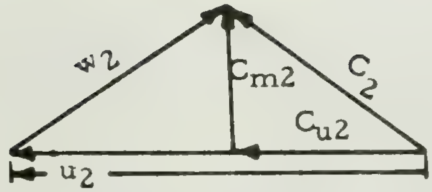


Fig. 3-2 Discharge Velocity Triangle

c. Design Elements Peculiar to the Pump Types

(1) Single Stage vs. Multi-Stage

The single-stage (three-lift) pump of Byron Jackson differs from the other two pump types, in that the impeller is "overhung" and the drive shaft does not pass through the inlet. As a consequence, there is more space in the inlet region and the general shape of the impeller is a bit different. The multi-stage pump impellers must accommodate the shaft and, therefore, have relatively greater inlet diameters, D_1.

(2) Multi-Stage Pump — Stage Variations

When drawings were received from BLH/Voith and A-C/Sulzer, it was discovered that both of their pumps had variation from one stage to another. Original calculation of specific speed, etc., were based on the assumption that the head per stage was the same on all stages. With variation in stage design, this is not strictly true. However, for rating the specific speed of the pumps, the stage head is still assumed to be the total head/number of stages. Treatment of internal design quantities is explained below:

(a) BLH/Voith — Two-Stage - Double Flow

The two single-flow first stage impellers have a larger eye than the second stage double-flow impeller. This is a realistic approach to improving first stage cavitation conditions while providing optimum hydraulic and mechanical design for the diffusers and second stage. Thus, inlet conditions differ from first to second stages.

The overall diameter of all the impellers is the same. Thus, the head per stage can be assumed equal, even though the impellers are not quite geometrically similar. The figures given in Table 3-I show the differences in inlet quantities.

In the model test program, several impellers are to be tested, e.g., Impeller A, Impeller B, Impeller C, etc. The finally selected first and second stage impellers and diffuser designs may be significantly different than those used for this analysis.

(b) A-C/Sulzer — Four-Stage Pump

The first, second and third stage impellers are identical, but the fourth stage impeller is significantly different. It is larger in diameter and has a little different inlet configuration. The reasons for the fourth stage variance have not been explained as yet. In computing design numbers, the inlet differences have been taken into account and the head per stage has been

considered by splitting the total head into three equal portions, plus a larger portion that is based on the square of the impeller diameter ratio:

$$H_{total} = \left[3 + \left(\frac{D_2\,(4th\ stage)}{D_2\,(1st, 2nd, 3rd\ stage)}\right)^2\right] H_{1st,\,2nd,\,or\ 3rd\ stage}$$

and

$$H_{4th\ stage} = \left(\frac{D_2\ 4th}{D_2\ 1st}\right)^2 \times H_{1st\ stage}$$

The resultant values are given in Table 3-I.

Again, the design is subject to change during the testing of the model pumps and components.

2. Evaluation of the Designs

a. Basic Design Coefficients

The character of the pumps under consideration can quickly be exposed by comparing their design parameters with "average" design constants and coefficients. Borrowing from Stepanoff, FIG. 3-3, 3-4 and 3-5 present "average" values of the design parameters. FIG. 3-3 gives "K" constants for geometric and velocity relations as a function of specific speed. (See the notation and definitions given previously in 1.b of this chapter). Similar to the "K" constants, the head and flow coefficients, ψ and ϕ, can be used for design guides. "Practical values are given in FIG. 3-4 and the general design spectrum is shown in FIG. 3-5. These constants and the coefficinets are by no means absolute in design use, but they are excellent guides. The one very critical element missing from this analysis is the impeller blade angle. "Average practice" generally produces a discharge angle of 22-1/2 degrees.

Calculated values for K's and ψ's and ϕ's etc., for the three Tehachapi pump types are given in Table 3-I . Average values from Stepanoff's data are given in brackets. ψ and ϕ values for the three Tehachapi designs are shown on FIG. 3-5. Comparison of design numbers leads to the following comments:

(1) ψ and ϕ values for all three pumps are normal except as noted in paragraph (2) which follows.

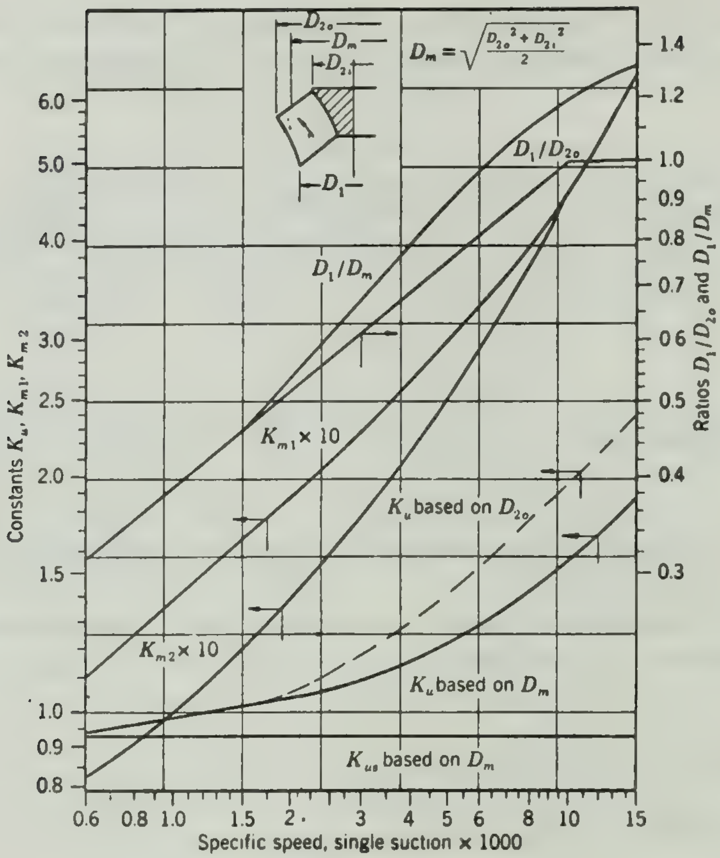


FIG. 3-3 STEPANOFF'S "AVERAGE, NORMAL" IMPELLER CONSTANTS

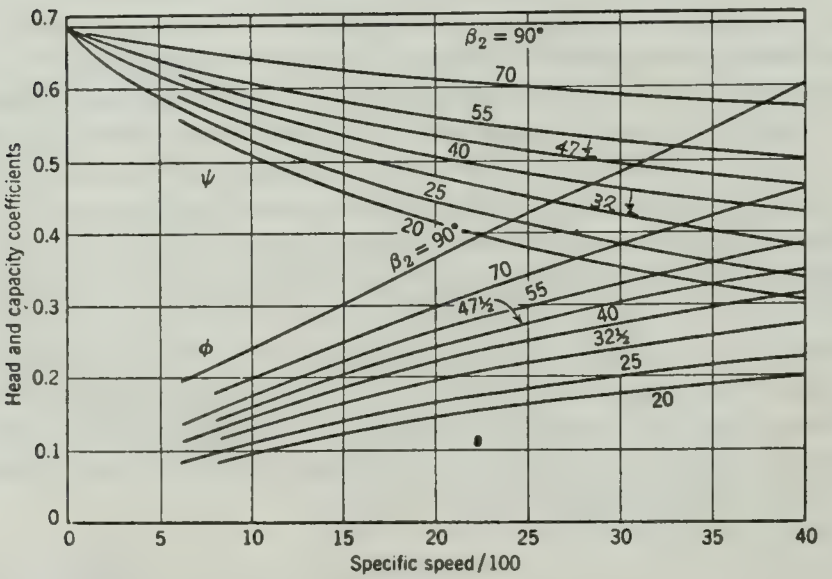
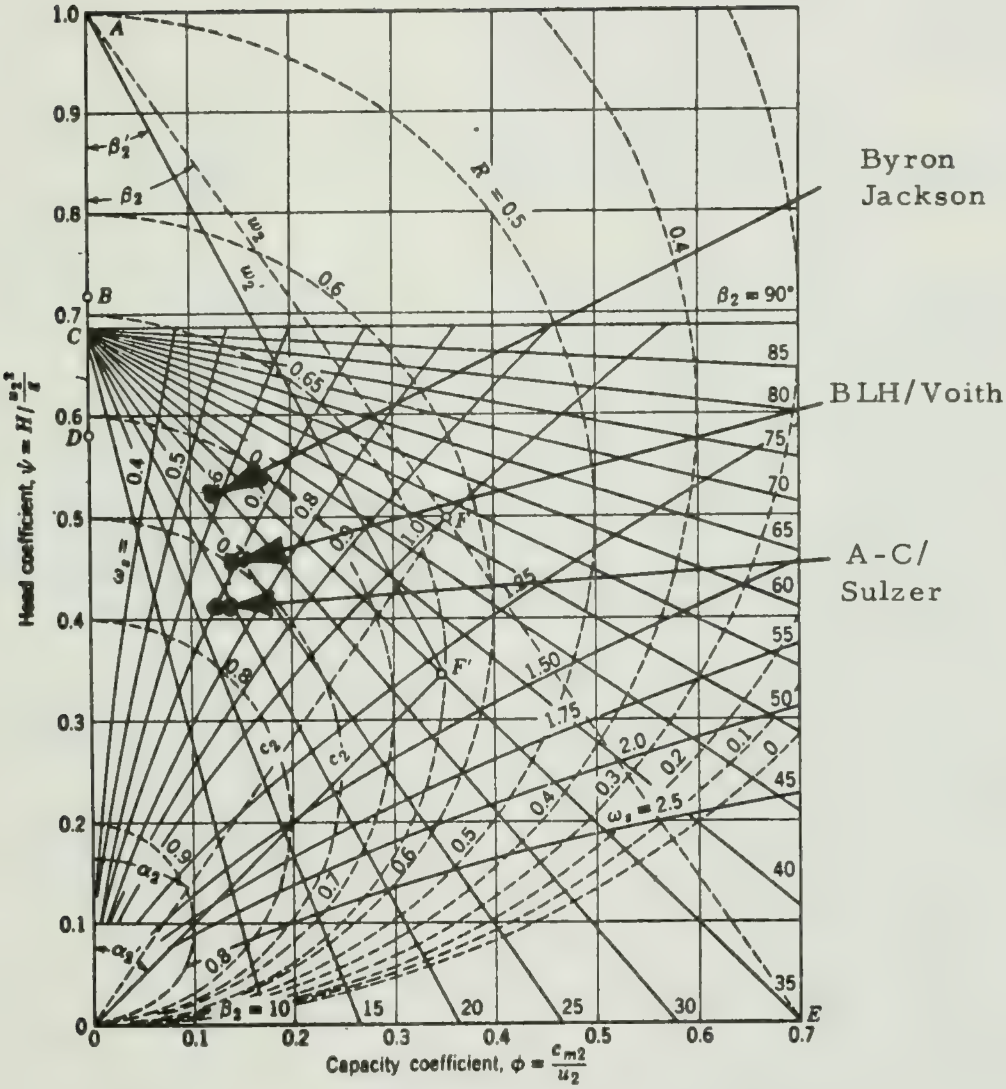


FIG. 3-4 STEPANOFF'S "PRACTICAL" VALUES OF ψ AND ϕ



Author's chart of centrifugal and axial pump characteristics.

FIG. 3-5 MODEL TEST FIRM
PREDICTED DESIGN COEFFICIENTS SHOWN ON
STEPANOFF'S "AUTHOR'S CHART"

(2) The ψ value for the Byron Jackson pump is higher than normally encountered. The design practice leading to this value is well founded on the Grand Coulee design and, therefore, must be accepted. Referring to FIG. 3-4 and 3-5, the B-J ψ value probably results to some degree from the use of a greater vane angle than used by BLH/Voith or A-C/Sulzer. The ψ coefficient of FIG. 3-4 would indicate a very high blade angle for B-J. However, the B-J design is simply different from "average" because the B-J discharge angle is 24° and 27° respectively for two experimental model impellers.

For discharge blade angles between 20° and 25°, the ψ and ϕ values of FIG. 3-4 are reasonably close to both the BLH/Voith and A-C/Sulzer designs.

(3) The K_u value is related to ψ ($\psi = \frac{1}{2K_u^2}$) and tells essentially the same story.

(4) K_{m_1} values for all three pumps for the inlet stage are very close to common practice.

(5) K_{m_2} values are reasonably close to common practice.

(6) Diameter ratios (D_1/D_2) are somewhat high for all pump types as compared to Stepanoff's recommendations.

(7) These "average" design parameters represent a time range of experience and do not, necessarily, represent most modern trends toward improvement. Therefore, where deviations occur they must be viewed from the aspects of the particular design problem and the result the deviation will produce.

b. Absolute Design Values

Study of the dimensional quantities and absolute velocities in the pumps (Table 3-I) shows that they are all very similar. Specific comparisons follow:

(1) The impeller overall and inlet diameters and discharge and inlet shroud spacing (D_2, D_1, b_2 and d_1) are very nearly the same for the A-C/Sulzer and BLH/Voith designs. The B-J design is larger overall because it turns at a slower speed and produces a higher head per stage. (Note: It must be remembered that the BLH/Voith design is double flow with two first-stage impellers handling one half of the flow and with a "double" second stage impeller with two inlets).

(2) The eye areas and discharge areas are likewise similar.

(3) The maximum tangential vane velocity at the vane inlet, u_1 varies from 114.1 ft./sec. for B-J to 128.7 ft./sec. for A-C/Sulzer with BLH/Voith at 120.9 ft./sec. (Inlet stages). The shaft-through-eye design of the latter two pumps requires the higher vane velocity.

(4) The meridional fluid velocity at the impeller eye, C_{m_1}, is very nearly the same for all pumps, 33.4 to 37.9 ft./sec.

(5) The meridional discharge velocity C_{m_2} is comparable for all three pumps, 24.85 to 27.35 ft./sec.

(6) The impeller tip speeds vary from 185 to 200 ft./sec.

c. Conclusions Concerning Hydraulic Design of Impellers

(1) All three designs follow conventional hydraulic design practice and are reasonably similar to each other. The A-C/Sulzer and BLH/Voith designs closely follow the design practice expressed in A. J. Stepanoff's text book, Centrifugal and Axial Flow Pumps. The Byron Jackson design deviates from that text book's recommendation, but is well founded on B-J's past practice, as exemplified by the Grand Coulee pump.

(2) The hydraulic performance predicted by each firm should be readily achieved.

(3) The reliability of the pumps is a function of their mechanical design. The hydraulic designs offer no unusual flow circumstance that would affect mechanical integrity.

d. Note on Multistage Diffuser Design

The design of the first stage diffuser and return channel in the Voith pump is different than the design of the diffuser-return channels of the Sulzer four stage pump. Referring to the cross-sectional drawings given in Chapter 6 and 7 in Volume I, the following differences can be noted.

The Voith diffuser has relatively long vanes and permits a considerable conversion of velocity head to static head before the flow is turned back to the next stage. The fluid velocities, therefore, will be low in the turn. The turn section does not have vanes and the return channel has a separate set of vanes straightening the flow for entry to the second stage.

In the Sulzer pump, the diffuser, turn, and return channels have continuous vanes. The turn occurs very close to the impeller discharge. As a consequence, the fluid velocity is still fairly high when making the turn and is subject to more violent secondary flow. The advantage of this design is that the pump case diameter is kept to a minimum. However, it may not be an optimum hydraulic design.

The subject of optimum diffuser-return design could be debated extensively, however, it is possible that the four-stage pump might achieve a higher efficiency if the diffuser-return design was more like that of Voith. There are other design differences in the Voith and Sulzer pumps that affect efficiency such as the balancing labyrinth on the four-stage pump and the double flow second stage impeller on the second-stage pump. These items effect friction losses. However, if these items are set aside, then the pumps could be almost hydraulically identical. They have nearly the same specific speed —the difference being due to the slightly greater flow in the Sulzer inlet (the Voith double flow pump has two inlets dividing the total flow). They have the same rotating speed and the same head per stage. Therefore, they could conceivably have nearly the same efficiency.

Further support to this argument can be derived by checking the ψ and ϕ values of the two designs. Referring to FIG. 3-5, Stepanoff's chart, the Voith design has a higher head coefficient, ψ , than the Sulzer design even though they have almost the same values of flow coefficient, ϕ . The Voith design ψ and ϕ values more nearly coincide with general experience. The Sulzer values must be due to the use of a lower than common blade angle B_2 (less than 20°) or else the lower value of ψ represents a performance penalty due to excess losses in the diffuser.

It will be interesting to compare the single stage model performances of Voith and Sulzer.[1]

3. Specific Speed Relationship

a. General

All three pump designs have specific speeds of about 2,000. This is a result of the performance conditions H, Q, and N specified for the pump development. As the question of optimum or correct choice of specific speed has been raised a number of times, a discussion of this parameter must be given.

b. Discussion

(1) Explanation of Specific Speed

Specific Speed is defined as: $N_s = \dfrac{NQ^{1/2}}{H^{3/4}}$ and in American

[1] See the model efficiency report in Chapter 9, Volume I and the recommendatio of Chapter 2, Volume I.

practice Q is given in gallons per minute and H in feet of fluid. Q is the pump flow rate through the inlet which will be one half of the pump total Q for a double-flow pump. H is the head per stage. Both the H and Q values are taken at the best efficiency point (b.e.p.) of the pump performance curve. A calculation of specific speed, using off b.e.p. values would have no real meaning. N_s is not a dimensionless number. The value of N_s gives an indication of the shape of the pump impeller as shown in FIG. 3-6. Also shown in FIG. 3-6 is the general relation of efficiency to Specific Speed. This information was developed by the Worthington Corporation and is widely quoted in American text books on hydraulics and pump design.

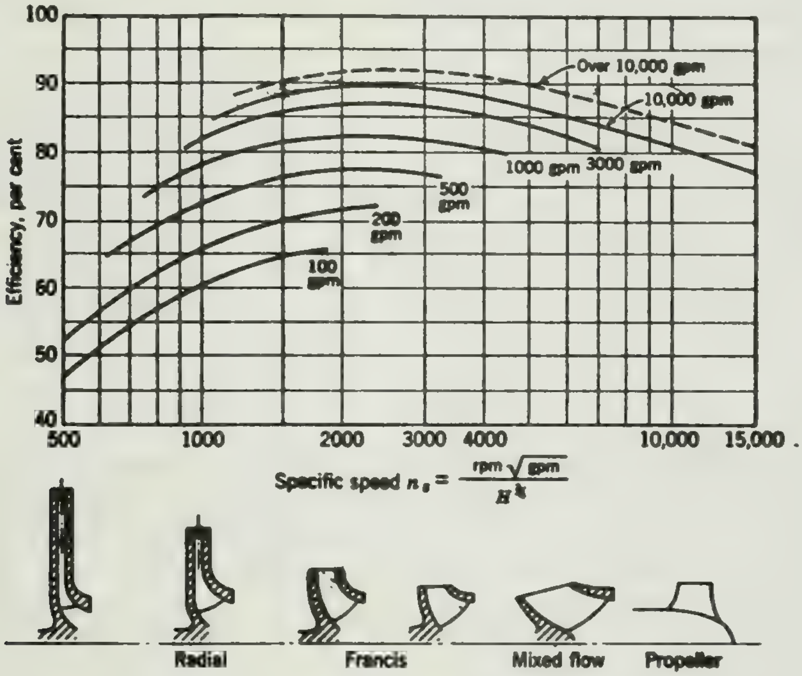


FIG. 3-6 SPECIFIC SPEED, SHAPE, SIZE & EFFICIENCY RELATIONS FROM WORTHINGTON

The specific speed does not tell the absolute rotating speed of a pump. It does not tell the number of stages in a pump. It does not tell the physical size of a pump. A pump with an impeller an inch in diameter and turning 50,000 RPM may have the same specific speed as a pump with an impeller several feet in diameter and rotating a few hundred RPM. Although the shape of the impeller does influence the mechanical design, the quality of the mechanical design is not related to specific speed and the whole question of reliability is, at best, only very remotely a function of specific speed.

It has been suggested, however, that pumps with specific speeds around 1500 are more reliable than pumps with specific speeds of 2000. This idea apparently comes from the observation that many large pump-turbine plants have pumps in the lower specific speed range. To demonstrate the fallacy in this line of thinking, consider the following comparison of pump types that are possible solutions to the Tehachapi lift:

Pump "A" for a two-lift system -- single stage -- 975 feet of head.

Pump "B" for a three-lift system -- single stage -- 650 feet of head. Both pumps will have the same flow rate and same rotational speed. Using H, Q, and N to calculate specific speed, the pumps can be compared in tabular form.

	PUMP "A"	PUMP "B"
N =	514 RPM	514 RPM
Q =	555 CFS	555 CFS
H =	975 FT	650 FT
N_s =	1480	2000

Pump "A" has the lower specific speed, yet it is identical to Pump "B" except it has <u>higher</u> head. Pump "A" with the higher head should have a greater wear rate with its wear rings. In order to produce the required head, "A" would have to be larger in diameter than Pump "B" and would require greater horsepower. It would appear then that "A" would be the least reliable of the two, therefore, lower specific speed cannot favor greater reliability.

(2) Efficiency Relationship

The Worthington curve shows that large pumps develop the best efficiency between the specific speeds of 2, 000 and 3, 000. Good efficiencies may be found in individual pumps at relatively low and relatively high specific speeds. FIG. 3-6 represents the trend found from a very large number of efficiency tests.

There is basic reasoning to the optimum specific speed - efficiency relation. Low specific speed pumps have long, narrow hydraulic passages and large friction surfaces that result in friction losses that are a high percentage of the total pumping energy. Leakage losses are also a high percentage of the total flow. In high specific speed pumps, relative velocities between fluid and pump surfaces are high in relation to the head being generated, and the ratio of friction loss to total pumping energy tends to increase as values of N_s go higher. Consequently, there is an optimum specific speed, even though the range of numbers is fairly broad. Many collections of pump performance data support the optimum specific speed values as given typically in FIG. 3-6. In a textbook by R. L. Daugherty,[1] an optimum curve is given which shows maximum efficiency at N_s = 2,500. The curve was supplied by the Byron Jackson Co. — a selected model test firm for the Tehachapi study.

c. Model Test Firm Position on Specific Speed

In the process of selecting the optimum specific speed for the Tehachapi models, the experience of the model test firms was sought and utilized. In support of the specific speed selection, a report was prepared by DMJM in October 1964 entitled, "Efficiency Specific Speed Relations." This report included submittals by the model test firms and the body of this report is again presented here.

[1] "Hydraulics" by R. L. Daughty, Fourth Edition, McGraw-Hill, New York, 1937.

(DMJM OCTOBER 1964 REPORT)

"REPORT ON

EFFICIENCY-SPECIFIC SPEED RELATIONSHIPS

"1. INTRODUCTION

The argument has been presented that European experience indicates that a specific speed in the range of 1400 to 1600 results in pump efficiencies at least as good as is obtained at a specific speed of approximately 2000.

This report discusses the information obtained by Daniel, Mann, Johnson, and Mendenhail on the recent European inspection trip from plant visits and from pump manufacturers and also includes a discussion of United States practice.

"2. THEORETICAL CONSIDERATION

Theoretical considerations certainly indicate the validity of the advantages of the higher specific speeds for best overall efficiency. It must be realized, however, that pump design is still far from an exact science where every item in the flow path is subject to known laws and there will be honest disagreement among designers on this subject.

However, as a basic criteria, it can certainly be stated that a low specific speed impeller, with its larger wetted surface area and the resultant increased disc friction, is less efficient than the one having higher specific speed. It is exactly for this reason that one manufacturer is now proposing aeration of the impeller shrouds to improve the efficiency and the subject has been discussed in a previous report dated Feb. 4, 1964.

"3. DATA FROM EUROPEAN INSPECTION TRIP

The information obtained during the plant visits and from manufacturers on the pump efficiencies of the various units is shown in Fig. I. {FIG. I of original report deleted -- see Plates I and II in the preceding Chapter 2}.

It has already been pointed out in the Interim Report covering these visits, in general the higher specific speed units are the ones installed more recently although there are exceptions. This is in part due to the fact that the operating speed of a storage pump installation is not necessarily selected for optimum pump operation but may be picked to favor the turbine operation.

The information shown on {Plate I — Chapter 2} is, therefore, not conclusive, although the average definitely shows a trend of increasing efficiency with increase in specific speed.

"4. DATA FROM MODEL TEST FIRMS

The three manufacturers which are under contract with Daniel, Mann, Johnson, and Mendenhall for building models were contacted in regard to this question and their answers are attached in the Appendix of this report -- following.

Sulzer Bros./Allis Chalmers and J. M. Voith are very definite in their statements that a specific speed in the range selected should be chosen. Byron Jackson Pumps, Inc., is a little less definite, giving a range of 1800 to 2400 as the best value for this application.

"5. REMARKS ON UNITED STATES PRACTICE

To throw some further light on the subject recent American practice on some large pumping installations is outlined in the table on the next page.

LIST OF U. S. PUMPING PLANTS
WITH PRINCIPAL OPERATING DATA

Name of Plant	Head FT.	Cap CFS	Speed RPM	Spec. Speed	Field Efficiency	Manufacturer	Operating Since
Iron Mountain	146	204	300	2160	91.4	Allis Chalmers	1941
Tracy	197	767	180	2000	91.6	Worthington	1952
Grand Coulee	310	1350	200	2100	93.5	Byron Jackson	1951
Flat Iron	240	370	300	2000	91.0	Allis Chalmers	1953

Here are three major United States manufacturers that have all selected a specific speed of 2,000 and above for some of their large installations. It would seem, therefore, that there is no question that for an installation where high efficiency is of primary importance, a specific speed of approximately 2,000 is the one that would be selected by American manufacturers."

"6. CONCLUSIONS

Based on the authoritative information contained in this report, there should be no doubt whatever that the specific speed selected for the State's model testing program is in accordance with the latest up-to-date practice both in the United States and in Europe.

APPENDIX TO DMJM OCTOBER 1964 REPORT

REMARKS BY MODEL TEST FIRMS

APPENDIX

SECTION 1. ALLIS CHALMERS MFG. CO.

Allis Chalmers Mfg. Co. (Sulzer Bros.) in their letter of October 7, 1964, commented as follows in regards to the 4-stage model pump in the Tehachapi program:

"This has reference to your letter of September 25, 1964 requesting information regarding efficiency values at various specific speeds."

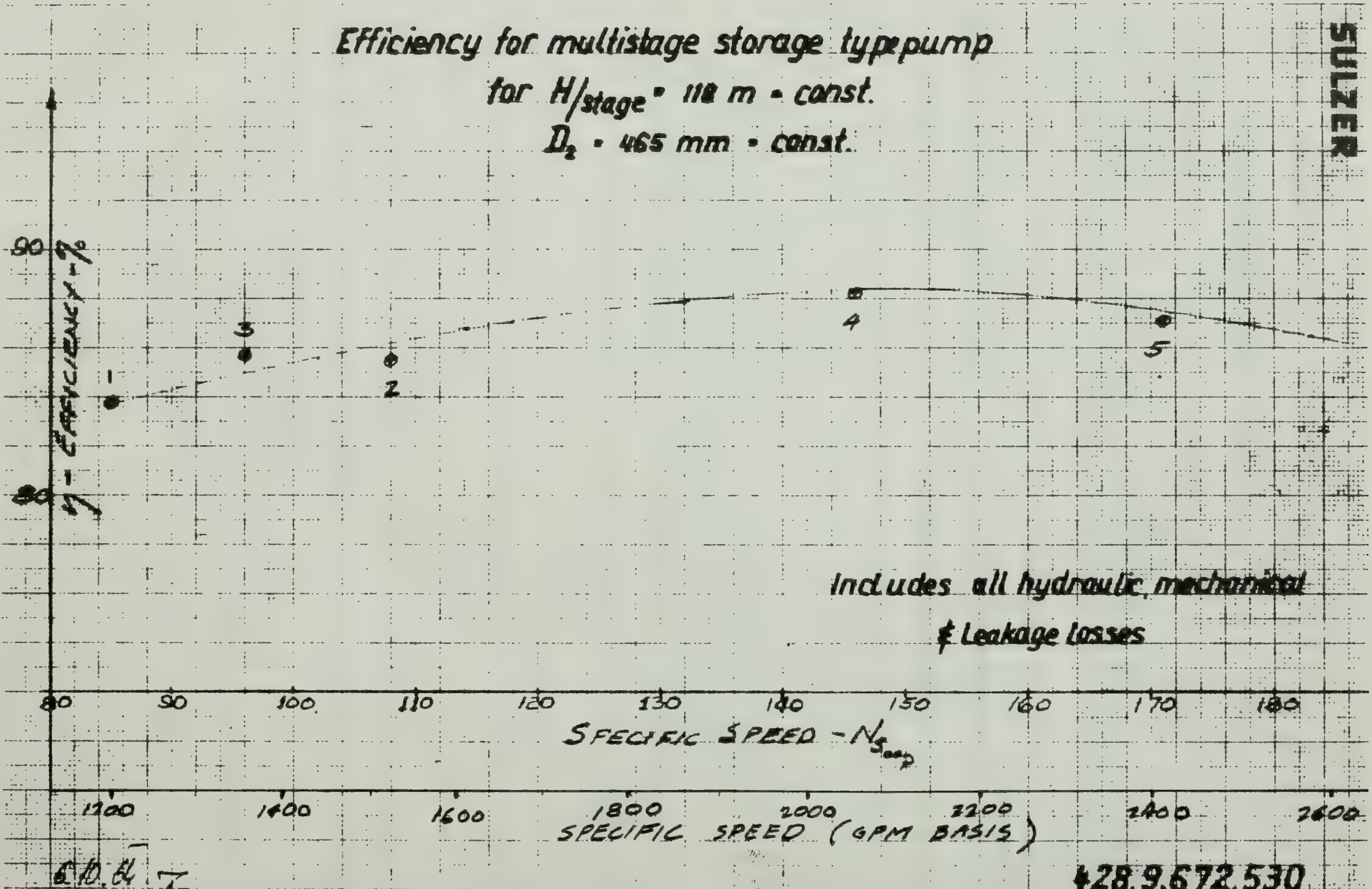
"You will find enclosed calculation sheets numbered 1 to 3 and a copy of curve 428.9.672.530 and 428.9.120.063. They have been prepared as follows: Data for five models of different specific speeds was tabulated. Models No. 4 and 5 did not have balancing labyrinths and the efficiency has been corrected to compensate for this. Then all the model results were converted to an equal basis - i.e. to a constant head and constant size. The constant values used were the average of all the model heads and sizes. The majoration equation was used for the conversion. The efficiency values vs. N_s of the 'common' or 'constant' model were plotted on curve 428.9.672.530. Using this curve of model efficiency, the prototype efficiency was calculated for various solutions (4 stage) to the Tehachapi Pumping Plant. These efficiencies have been plotted (vs. N_s) on curve 428.9.120.063."

"It is quite clear that an N_s of 148 to 158 is the best range. This equivalent to 2090 - 2230 in English specific speed."

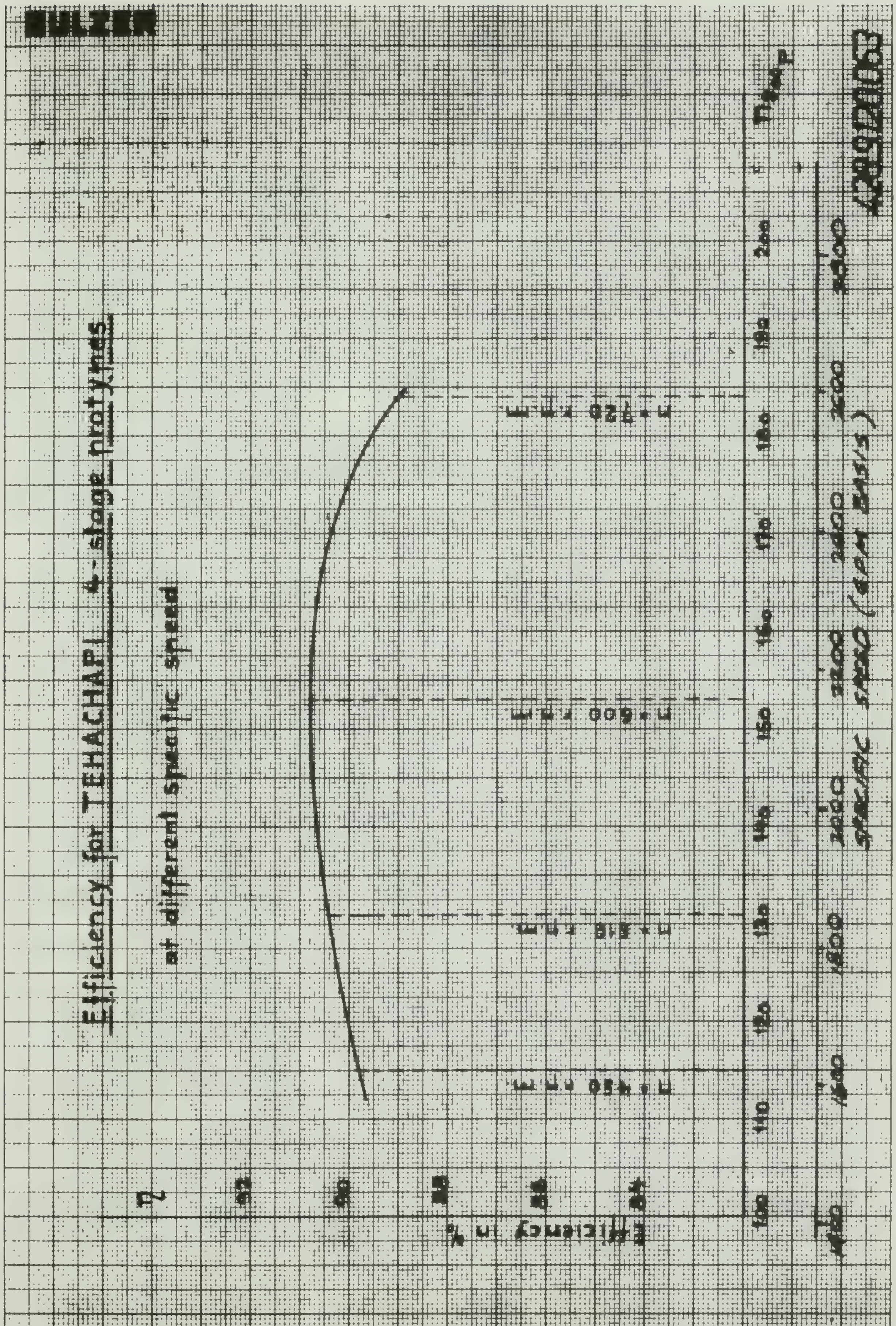
In addition, Allis Chalmers Mfg. Co., in their letter of October 1, 1964, gave the following comments pertaining to single stage pumps.

"In general, I believe one of the best comparisons of specific speed and efficiency is on the Metropolitan aqueduct pumps. There were pumps of three different specific speeds consistent with the operating head and suction conditions. The specific speed of about 2000 was used on the Iron Mountain pumps which we manufactured and was the highest of all the pumps on the aqueduct by a considerable margin."

Material prepared for this spot falls within the category of information considered proprietary by Daniel, Mann, Johnson, and Mendenhall and has been deleted at the request of that firm.


SULZER
Efficiency for multistage storage typepump
for H/stage = 118 m = const.
D2 = 465 mm = const.
η - EFFICIENCY - %
90
80
1
2
3
4
5
Includes all hydraulic, mechanical
& Leakage losses
80
90
100
110
120
130
140
150
160
170
180
SPECIFIC SPEED - Ns
1200
1400
1600
1800
2000
2200
2400
2600
SPECIFIC SPEED (GPM BASIS)
428.9.672.530

MODEL TESTS



SULZER
Efficiency for TEHACHAPI 4-stage protypes
at different specific speed
η
92
90
88
86
84
Efficiency in %
n = 450 r.p.m
n = 518 r.p.m
n = 600 r.p.m
n = 720 r.p.m
100
110
120
130
140
150
160
170
180
190
200
1400
1600
1800
2000
2200
2400
2600
2800
SPECIFIC SPEED (GPM BASIS)

J. M. VOITH GMBH · HEIDENHEIM (BRENZ)

MASCHINENFABRIK

Fernsprecher: Nr. 3221
Durchwahl Nr. 322..........
Ortskennzahl: 07321
Fernschreiber: Nr. 7-14866
Telegramme: Voithwerk Heidenheimbrenz

Baldwin-Lima-Hamilton
Corporation

Philadelphia 42, Pa.

U.S.A.

Ihr Zeichen	Ihre Nachricht	Unsere Zeichen	7920 Heidenheim (Brenz)
		thv 561 DrDz 20040	October 7,1964 OMM/Mr.

OCT 1964

Betreff TEHACHAPI
Your Telex of Oct.1,1964
Our Letter of Sept.25,1964
Our Telex of Oct.6,1964

Gentlemen:

There is a distinct dependency of the efficiency of a centrifugal pump from the specific speed as shown already on our graph 2.82-4108. In the meantime, we have supplemented this graph by values which have been measured on model impellers of several projects (see enclosure). The curve can also readily be explained theoretically. For the same impeller diameter and the same speed, the absolute value of the losses on the impeller sides remain of the same order. However, the ratio of the loss on the impeller sides to the pump power falls off as the specific speed increases and so does the delivery and the power of the impeller. When the clearance losses are considered, the curves show a similar trend. In the literature similar efficiency curves have been published, e.g. fig.10.9 A.J.Stepanoff,1948. The maximum efficiencies published in this article lie between specific speeds of 2,000 and 3,000: thus, approximately+the same order as on our diagram. The permissible setting of the pump

+ in

Re: T E H A C H A P I

depends on the specific speed and the delivery head of the stage involved. In order to avoid deep settings, normally low specific speeds are favoured. This applies in particular to the existing plants in Europe, where only a few underground stations have been built. However, for the TEHACHAPI Project, the efficiency is of such importance that, in our opinion, a higher mean specific speed of the order of n_s = 1,300-2,300 should be chosen.

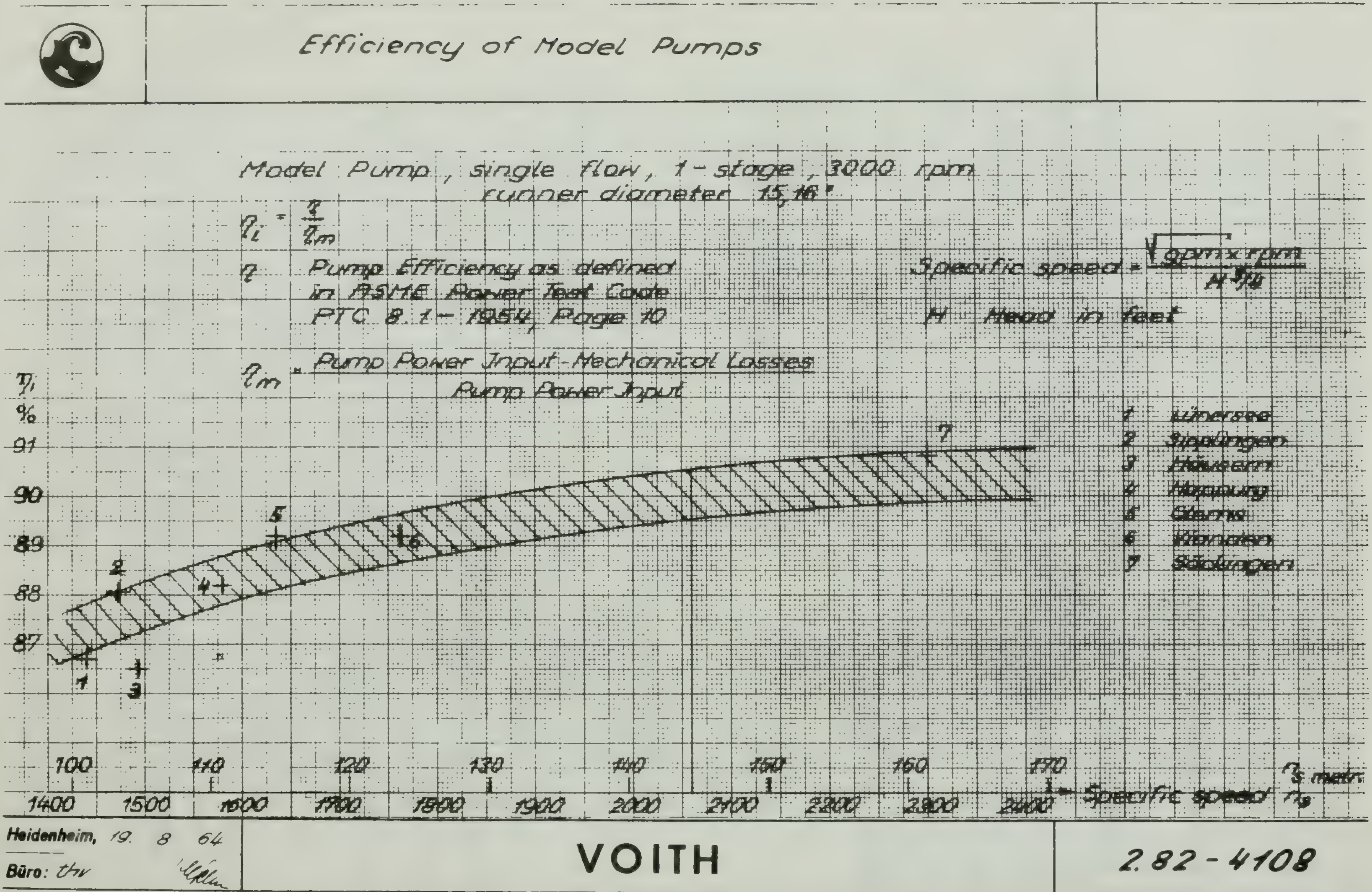
Yours very truly,
J.M. VOITH G.BH

Reviewed and approved by: ____________________

For : ________________________________

Date: ________________________________

Enclosure



Efficiency of Model Pumps
Model Pump, single flow, 1-stage, 3000 rpm
runner diameter 15,16"
$\eta_i = \frac{\eta}{\eta_m}$
η Pump Efficiency as defined in ASME Power Test Code PTC 8.1-1954, Page 10
$\eta_m = \frac{\text{Pump Power Input} - \text{Mechanical Losses}}{\text{Pump Power Input}}$
Specific speed = $\frac{\sqrt{gpm} \times rpm}{H^{3/4}}$
H Head in feet
η_i
%
91
90
89
88
87
1 Lünersee
3 Häusern
4 Happurg
5 Glems
7 Säckingen
100
110
120
130
140
150
160
170
n_s metr.
1400
1500
1600
1700
1800
1900
2000
2100
2200
2300
2400
Specific speed n_s
Heidenheim, 19. 8. 64
Büro: thr
VOITH
2.82-4108

BYRON JACKSON PUMPS, INC.
A SUBSIDIARY OF BORG-WARNER CORPORATION
2301 East Vernon Avenue, Vernon, California • Mailing Address: P. O. Box 2017, Terminal Annex, Los Angeles, California 90054 • LUdlow 7-6171

October 12, 1964

Daniel, Mann, Johnson
& Mendenhall
3325 Wilshire Boulevard
Los Angeles, California

Attention: Mr. Hans Gartmann
Project Engineer

Subject: Tehachapi Pumping Facility
Specification No. 637-1-1C
Efficiency vs Specific Speed
Your Letter September 25, 1964

Gentlemen:

The correlation of pump efficiency to specific speed covers a highly complex field, due to the many variables involved. Research data available in this field, including our own, is of a general nature, covers mainly single stage pumps, and tests are performed on sizes where surface finishes and commercial manufacturing methods may distort the picture to a considerable extent.

Furthermore, test facilities of an acceptable capacity and of an accuracy which could establish an ultimate specific speed have only very recently been developed. This makes present research data suspect.

With the many variables, it is our opinion that an ultimate specific speed range for optimum efficiency exists, rather than a single specific speed value. Our data shows this range to lie between 1800 and 3500 specific speed.

The main variables which would determine a closer band within this range are:

1. Absolute pump size.
2. Number of stages.
3. Case design (single volute, double volute, diffuser volute, straight diffuser).
4. Practical manufacturing methods available.

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The general effect of these variables is as follows:

A. For a small absolute pump size a specific speed in the upper end of the range can be expected to produce the best efficiency. This is due to the effect of friction losses.

B. For multi-stage pumps a specific speed in the lower range is most likely to produce the best efficiency. This is due to leakage losses and friction losses in cross-overs.

C. The volute design favors the lower and middle specific speed range, while the diffuser favors the middle and upper specific speed range. The effect stems from surface friction and induced turbulence. Our data indicate both ranges to be quite wide.

D. Where complexity of design and manufacturing methods prevents good surface finishes, the best efficiency will be obtained in the upper specific speed range.

For the Tehachapi Pumping Facility we believe a specific speed range of 1800 to 2400 will result in the best efficiency for the three (3) pump types under consideration. Only extensive testing could determine the effect on the efficiency outside this range.

Very truly yours,

BYRON JACKSON PUMPS, INC.

E. B. Fiske

E. B. Fiske
Senior Engineer

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d. Effect of N_s on Cavitation Performance

Looking at the definition of specific speed, $N_s = \frac{NQ^{1/2}}{H^{3/4}}$ it can be seen that if H is fixed by a static lift and a choice of total pumping stages to perform that lift (pump stages x pump stations) then a higher specific speed pump must be designed for higher rotating speed, N, and/or higher flow, Q . Also, looking at the definition of the cavitation performance indicator, suction specific speed,

$$S = \frac{N Q^{1/2}}{(NPSH)^{3/4}}$$

it can be deduced that to maintain a specified S limit, the NPSH must be higher for a higher specific speed pump. Thus, for the Tehachapi application a high specific speed pump will require a greater NPSH and consequent greater submergence than a low specific speed pump.

The optimum specific speed for efficiency could thus be compromised to obtain a lesser submergence. The economics of cost of a point of efficiency versus cost of excavating for increased submergence must be studied. For the Tehachapi application, the power level is very high and a long operating life is anticipated and so the pump efficiency dominates the economic situation.

e. Influence of N_s on the Number of Stages

With a completely fresh design problem a pump engineer will consider various values of N, Q, H, S, and horsepower for a single pump. The total Q for the application may be divided among several pumps in parallel in order to meet horsepower or submergence limitations. Rarely is there an infinite choice of rotating speeds, and certain definite speeds must be considered. The total head may be split into several pumping stations or may be split between several stages of a multistage pump. In the process of isolating choices, certain trial values will be used for checking design feasibility and it is common to first check the specific speed. If the specific speed is very low, then the efficiency will be poor and improvement can be made by going to two or more stages. N or Q may also be adjusted to obtain a more nearly optimum specific speed. There are many factors that may influence the final design but in general, multi-staging is used for "specific speed" improvement.

The tendency in multi-stage pumps will be to use the minimum number of stages as required to obtain a reasonably good specific speed — efficiency arrangement. Where pumps are used for pump storage operation,

turbine operation may compromise the pump design and/or the intermittent character of the operation may reduce the importance of efficiency in the plant design considerations. For these reasons, pump-storage units may employ lower specific speeds than what would be considered optimum for a high efficiency aqueduct application. The reliability of the pumps is an independent consideration.

C. MECHANICAL DESIGN

1. General

A major consideration in the choice of the model test firms was their individual experience with the design and manufacture of the pump types to be studied. Thus, the mechanical design of each pump should be the most advanced of its type. Individual description of the pumps are given in Volume I, Chapters 6, 7 and 8, along with sectional drawings. Many mechanical features are discussed there.

DMJM has reviewed the preliminary prototype designs and finds them to follow the traditional approach of each of the companies. There are certain basic differences in the three designs as illustrated in Table 3-II where pertinent mechanical features are given.

It is expected that a thorough mechanical design analysis will be made when a particular pump type has been selected and interface data firmly established. The preliminary designs permit comparison of pump types. Each has certain advantages and disadvantages peculiar to the hydraulic type. The single-stage, three-lift pump is by far the simplest in appearance but arguments as to its greater reliability must also include consideration of the number of pumps required, the rates of wear of components, etc. The subject of reliability is thoroughly treated in Chapter 7 (Volume II).

2. Service in Plant

The single-stage pump will be the most easily serviced type. Removal of the head cover and bearing permits the removal of the single impeller or the front cover may be removed and the impeller can be removed from the inlet side. The two-stage double-flow pump must have the lower suction impeller removed from below and the upper impeller, double-flow impeller and diffuser sections removed from the top. Similarly, the four-stage pump requires working at both ends for disassembly and service, and has numerous internal sections. Both can be more readily serviced by removal of the entire pump. As a consequence, the servicing of either the two or four-stage pumps will be more involved than for the single-stage pump.

Table 3-II

DESIGN & CONSTRUCTION FEATURES OF THE TEHACHAPI PUMPS

Pump Type	A-C/SULZER four-stage		B-L-H/VOITH two-stage, double flow		BYRON JACKSON single-stage	
Mounting	Vertical		Vertical		Vertical	
Case	Cast sections welded together		Casting		Cast sections: scroll welded to diffuser section	
Impellers	Cast		Cast		Cast	
Wear Rings: Imp.	Separate ring	2-steps plus return step on each impeller inlet side	machined on impeller	3 steps plus return step on each impeller inlet	machined on impeller	grooved 'labyrinth' on both sides of impeller
Case	Separate insert		Separate insert		Separate insert	
Clearance	0.016 radial		0.024 radial		0 012 radial	
Seal Rings	2 step shaft sleeve		Straight path large insert in case shaft sleeve on top-plain shaft - on bottom		None	
Clearance	0.024 radial		0.018 radial			
Shaft Seals (Stuffing boxes)	2 seals; babbitt lined bronze segments with lantern ring and clear water supply		2 seals; sliding ring–lantern type with clean water supply		Grooved 'labyrinth' 0.012 radial clearance x 10" length with clean water introduced at center.	
Bearings	3 section adjustable: bearing on each end.		bearing on each end, no details given, yet.		single, self-aligning sleeve bearing(overhung impeller)	
Lubrication	Three options: 1) Self-lubricated-seal water cooled 2) Internal pressurization system (external source for starting required) 3) Complete external source.		No details given as yet.		External pressure source.	

3. Running Clearances and Seals

In the interest of achieving the highest possible pump efficiencies, manufacturers strive to reduce seal clearances and wear ring clearances as much as possible. It is not uncommon for plant operators to increase clearances when overhaul and/or repair work is done in order to overcome conditions where rubbing takes place, and to facilitate disassembly and assembly operations. Also, as the pump rings wear due to erosion, the clearances increase so that the average clearance over the life of the pump will be greater than the clearance provided in the initial installation. So, the pump efficiency will deteriorate in time until an overhaul is made and the pump is returned to near new condition. This fact must be borne in mind in aqueduct performance analysis.

A review of clearances specified in Table 3-II will show that very close fits are anticipated, especially in the case of the Byron Jackson single-stage pump. The time average running clearance and efficiency of each of the pump styles is difficult to predict, but the comparative ratios will probably be different from those when pumps are new. However, assignment of penalties for wear in the comparative selection process simply cannot be done fairly and the comparative analysis should consider model performance only.

It is interesting to note that BLH/Voith and A-C/Sulzer have opposite approaches to wear ring and interstage seal clearances. The four-stage pump employs 0.016 at the wear rings and 0.024 at the interstage seals, whereas, the two-stage double-flow pump uses 0.024 at the wear rings and 0.018 at the interstage seals.

The Byron Jackson single-stage pump must have reasonably close clearance on its labyrinth shaft seal in order to insure a reasonable limit on leakage. Assuming that materials with high erosion resistance are used in their design, the seal should give long service and eliminate the need for frequent servicing of seals and seal shaft sleeves with rubbing type seals. The elimination of rubbing seal friction is of direct benefit to overall pump efficiency providing the time average leakage loss is not so much greater than with other types of shaft seals as to cancel the gain.

D. CAVITATION PERFORMANCE

1. Introduction

Pump machinery is subject to cavitation problems of two kinds: (1) breakdown of performance due to excessive cavities in the hydraulic passages; and (2) wear and damage to pump parts due to the erosive action of cavitation. The first problem, that of performance deterioration, cannot be

tolerated. The second problem, wear, can be tolerated to a degree that is economically balanced by the cost of wear resistant materials for pump fabrication and repair costs on one hand, and the cost of installation requirements to reduce cavitation on the other. This study, when complete, will include results of model tests and will eventually conclude with recommendations for the amount of pump submergence, materials of construction for the prototype pump, and materials to be used in subsequent maintenance of the pumps.

2. General Discussion of Pump Cavitation

Centrifugal pumps are subject to cavitation in their inlet regions when operating with a low value of suction pressure such that the local static pressure on the surface of a vane or passage is reduced to approximately the vapor pressure of the liquid handled. Usually, only the impeller is involved, but sometimes inlet guide vanes or seal clearance regions in the pump may be subject to cavitation. The presence and degree of cavitation is thus dependent on the absolute pressure above vapor pressure in the pump suction and the value of the required minimum pressure is a function of the pump design and its operating speed. The quantity usually used to specify the pump suction requirements is the NET POSITIVE SUCTION HEAD. The abbreviation is NPSH. (See Section 3 following for a definition.)

A particular pump, operating at a fixed speed, will be completely free of cavitation if there is sufficient NPSH applied to its suction. If a reduced value of NPSH is employed, bubbles will form and flow from surface imperfections in the inlet regions of the impeller. This kind of cavitation may be responsible for some erosion, but it is generally not serious. With a further reduction in NPSH, extensive bubble formation and cavities will form on the low pressure regions of the pump impeller surfaces. As cavitation develops, bubbles form and are carried into the stream and then collapse in a region of higher pressure. Advanced cavitation consists of persistent attached cavities with bubbles streaming from the cavity tails. The area where cavities and bubbles collapse is where the erosion takes place. With well developed cavitation, erosion is more serious.

Considerable cavitation can be present in a pump without affecting hydraulic performance. However, when a relatively low value of NPSH occurs, then cavitation may be so prevalent as to modify the effective passage areas and flow patterns and cause a change in performance. With submerged pumps, the effect is to limit the head and flow that can be pumped. Pump performance is reduced both in head generating capability and in the efficiency of pumping. Operation with performance deterioration is accompanied by considerable noise and vibration and may result in pumping surges. When performance is so reduced, the term "cavitation breakdown" is employed and the value of NPSH for cavitation breakdown is generally determined by analysis or test and specified such that it may be avoided in operation.

The point where cavities first begin to form is termed "incipient cavitation" and the value of NPSH is much higher than for cavitation breakdown. The actual value of NPSH for incipient cavitation is hard to determine because the method of detecting incipient cavitation is not a simple matter. Sometimes it is determined acoustically (cavitation generates noise when bubbles collapse) and sometimes by visual observation. Occasionally, it is not detected until after long periods of pump operation when cavitation erosion is found during an overhaul or inspection.

The margin between incipient cavitation and cavitation breakdown also depends on the particular pump design and on the surface quality of the impeller. A design may have a region of cavitation that may advance considerably before affecting performance. Also, cavitation off surface irregularities generally has little effect on breakdown. Cavitation from surface imperfections must affect erosion, but the measure of this effect is not practical. Good surface finish is an asset, both for minimizing local cavitation and reducing friction losses.

In practice, manufacturers usually specify NPSH requirements based on breakdown, with some arbitrary margin of safety. Sometimes specified values of NPSH that will give minimum cavitation erosion are based on the manufacturers' experiences with field reports. Or, a value based on a code such as given by the Hydraulic Institute is employed.

The exactness of NPSH specifications have not been too good, however, because often an unsatisfactory rate of wear, due to cavitation erosion, has been found after a period of initial operation of the prototype machine(s). Eroded areas are then repaired, using materials with greater wear resistance, or impellers may eventually be replaced with a "stainless" material. Considering the size of the installation, and the reliability requirements for Tehachapi, a sophisticated study of cavitation, cavitation erosion and materials of fabrication will be valuable.

3. Cavitation Factors from the Operational Viewpoint

In pump literature, two symbols are commonly found for NET POSITIVE SUCTION HEAD. One is the abbreviation NPSH, and the other is: H_{sv} or h_{sv}. Various authors use primes, superscripts and subscripts for indicating breakdown, inception, etc.

NPSH is defined by:

$$NPSH = h_s + h_a - h_{vp} \tag{3-1}$$

where h_s = total head at the pump suction (suction head)

h_a = atmospheric pressure (head) = 34.0 ft for water at sea level

h_{vp} = vapor pressure (head) absolute = 1.2 ft for water at 80° F

The suction head, h_s, is measured relative to the pump centerline for horizontal pumps and at the "eye" for vertical pumps. At a pumping station with an open forebay, as in FIG. 3-7 h_s is the elevation of the water surface above the pump, less any friction losses occurring in the inlet piping up to the pump flange:

$$h_s = Z_s - h\ell_s \tag{3-2}$$

With short, well designed inlet piping, $h\ell_s$ may be neglected.

If the suction head is measured with a gage, then,

$$h_s = \frac{P_s}{w} + Z_g + \frac{V_s^2}{2g} \tag{3-3}$$

where P_s = pressure gage reading (static pressure).

Z_g = gage height above pump datum

V_s = mean velocity at pump suction = $\frac{Q}{A_s}$

A_s = area of pump suction.

w = specific weight of the fluid.

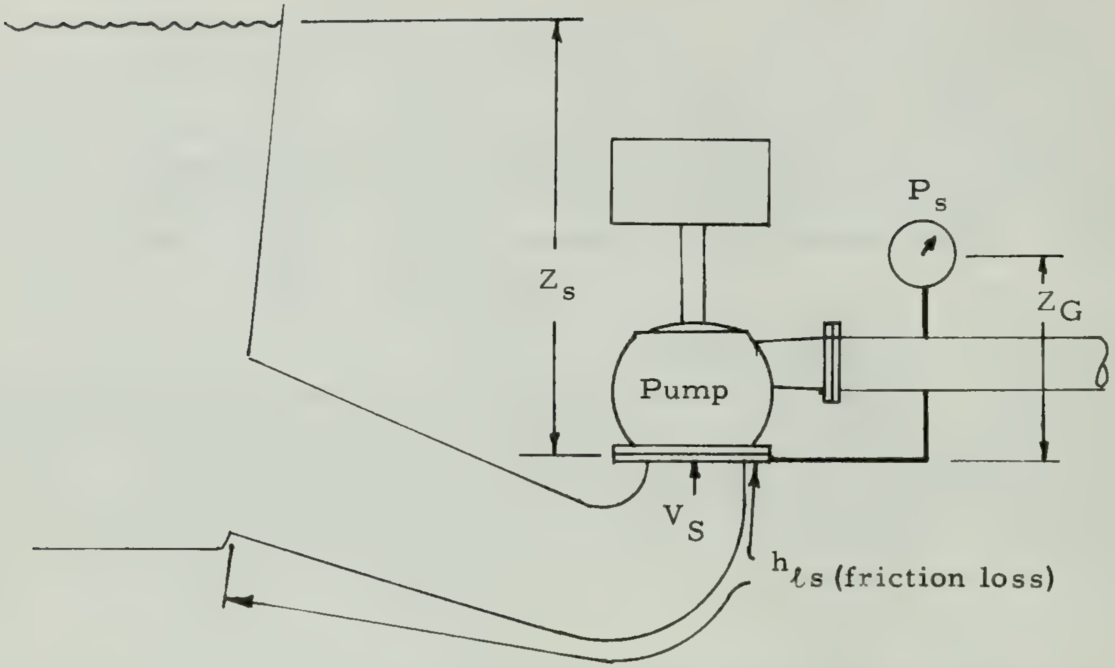


FIG. 3-7 SCHEMATIC OF PUMP SUBMERGENCE AND PRESSURE RELATIONS

When designing a pump installation, the required submergence, Z_S, can be calculated from a specified acceptable value of NPSH. The required NPSH can be calculated from experience for estimating use, but should be measured for absolute confidence by pump test or model pump test.

Testing a pump for cavitation performance is usually done by lowering the suction head while the pump is operating, until a change in performance is detected. A result as shown qualitatively in FIG. 3-8, is obtained. At NPSH values above breakdown, the pump will perform satisfactorily. At values of NPSH approaching the incipient value, cavitation erosion will be of little consequence. The value where cavitation erosion becomes serious is not generally known. The pump manufacturer may specify NPSH at breakdown as the required value, or he may specify an arbitrarily determined higher value.

In order to provide cavitation criteria that is suitable for evaluating the relative merits of various pumps and that is suitable for scaling, the dimensionless number σ is used for suction head where

$$\sigma = \frac{NPSH}{H} \tag{3-4}$$

H = pump head (per inlet stage).

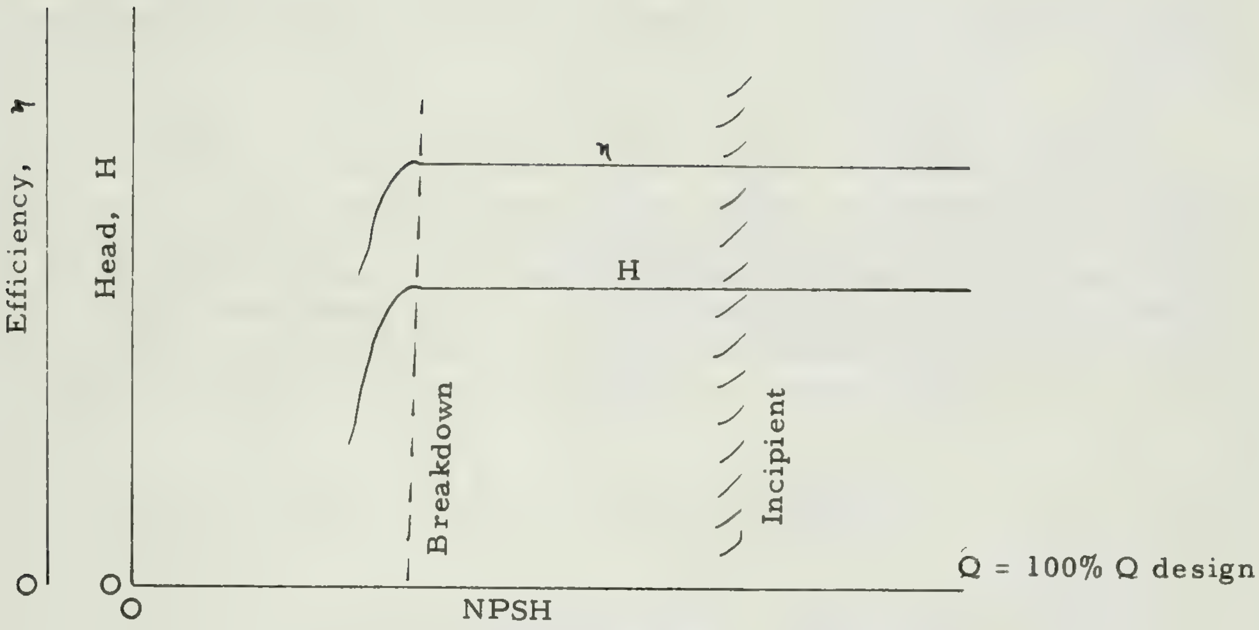


FIG. 3-8 TYPICAL CURVE OF CAVITATION PERFORMANCE TEST

σ is known as Thoma's cavitation parameter. A small value of σ is desired. Curves similar to FIG. 3-8, but with σ as the ordinate, can be obtained for various pump operating points and, so, curves of NPSH or σ can be prepared as functions of pump flow for use when pumps are to operate over a range of performance conditions. If a value of NPSH or σ is quoted without reference to an operating point, it is assumed to apply to the best efficiency point (or design flow rate). The σ quantity is widely used for specifying pump cavitation performance, but unfortunately its value is a function of the pump type. Another cavitation parameter known as Suction Specific Speed, S, is defined as:

$$S = \frac{N Q^{1/2}}{NPSH^{3/4}} \tag{3-5}$$

where N = pump RPM

Q = flow rate in GPM (use $^1/_2$ Q for double suction pump).

S is not dependent on the pump type; it is a measure of the inlet performance; and it is a scaling factor. It is desirable to achieve high values of S corresponding to low values for NPSH.

The Hydraulic Institute, in its 1961 review, offers several curves relating submergence (or suction lift), pump stage head, and Upper Limits of N_s for hot water, cold water, and for double suction, single suction with shaft through eye, and single suction overhung impellers. These curves converted to values of S give quite a scattering of recommendations. S varies slightly with N_s by these Hydraulic Institute recommendations. Older Hydraulic Institute recommendations gave a general value of S of about 7,900 and S = 8,000 is often quoted as a good "rule of thumb". The later Hydraulic Institute curves give:

a. S = approximately 8,600 for single suction impellers with shaft through eye and N_s = 2,000;

b. S = approximately 8,300 for double suction impellers with N_s = 2,000 (value based on 1/2 Q equivalent to single suction);

c. S = approximately 9,000 for single suction overhung with N_s = 2,000; all for water at 85° and sea level. Thus, the Hydraulic Institute would rate single suction overhung (c) as slightly better than "with the shaft through the eye".

In any event, it must be remembered that the Hydraulic Institute recommendations are "upper limits" and are based on breakdown and not on completely cavitation free operation where impeller surfaces would not suffer erosion. Pumps operating with S = 8,000 and even lower are found to suffer from cavitation erosion. Stainless steel impellers can be employed to minimize damage, but if wear is to be avoided for 50 years of operation, then conservative values of S (more submergence) should be considered. A value of S = 7,000 is tentatively recommended and has been used in making submergence calculations (see following section). If model tests (with visual inspection for cavitation) indicate a higher value of S is acceptable, then the change should be made in the final plant specification.

The value of S = 7,000 has been used for all pump types (lift concepts). The single suction overhung style of the three lift system will probably have cavitation performance that is somewhat better than the other two "shaft-through-eye" designs, but the difference will not be very large and special attempts to estimate the difference can better await model test results.

4. Submergence Calculations

Results of calculations using the equations previously developed, are summarized in Table 3-III .

The influence of S on the actual submergence is shown in FIG. 3-9.

The effect of a low water surface on cavitation conditions was investigated by calculating the resultant S for the minimum canal elevation. The result is to increase S from 7,000 to about 7,500-7,600, which is acceptable if the condition does not exist for too long a time. This condition of S is still below the Hydraulic Institute recommendation.

NOTE: σ is dependent on pump head and therefore is a function of specific speed. The relation between σ and Suction Specific Speed, S, is:

$$\sigma = \left(\frac{N_S}{S}\right)^{4/3} \quad \text{or} \quad S = \frac{N_S}{\sigma^{3/4}}$$

Table 3-III

NPSH & SUBMERGENCE CALCULATIONS FOR THE TEHACHAPI LIFT CONCEPTS

Model Firm		A-C/ Sulzer	BLH/ Voith	Byron Jackson
Lifts		1	2	3
Q cfs		312.5	277.8[1]	555.6
Q gpm		140,200	124,600	249,200
N rpm		600	600	514
$N\sqrt{Q}$		224,700	211,700	256,500
M.W.S. elevation	Plant 1	1,239	1,239	1,239
	Plant 2	–	2,203	1,881
	Plant 3	–	–	2,524
Atmospheric head, h_a	Plant 1 (ft)	32.5	32.5	32.5
	Plant 2	–	31.3	31.7
	Plant 3	–	–	31.0
$(h_a - h_v)$	Plant 1	31.3	31.3	31.3
(See footnote 2)	Plant 2	–	30.1	30.5
	Plant 3	–	–	29.8
NPSH FOR S = 8000		85	78	102
Submergence for:	Plant 1 (ft)	53.7	46.7	70.7
	Plant 2	–	47.9	71.5
(See footnote 3)	Plant 3	–	–	72.2
NPSH FOR S = 7500		93	86	111
Submergence for:	Plant 1 (ft)	61.7	54.7	79.7
	Plant 2		55.9	80.5
	Plant 3			81.2
NPSH FOR S = 7000		102.5	94.5	121
Submergence for:	Plant 1	71.2	63.2	89.7
	Plant 2	–	64.4	90.5
	Plant 3	–	–	91.2
Approx. Elevation of ℄ of Inlet stage for M.W.S. and for S = 7000	Plant 1	1,167.8	1,175.8	1,149.3
	Plant 2		2,138.6	1,790.5
	Plant 3			2,432.8
From elevations given above, the effect of a drop in the canal to the minimum water surface of 1229' at Plant 1 will be: Submergence		61.2	53.2	79.7
S		7,520	7,580	7,500

[1] Q total = 555.6; 1/2 Q total is used for a double suction pump calculation.

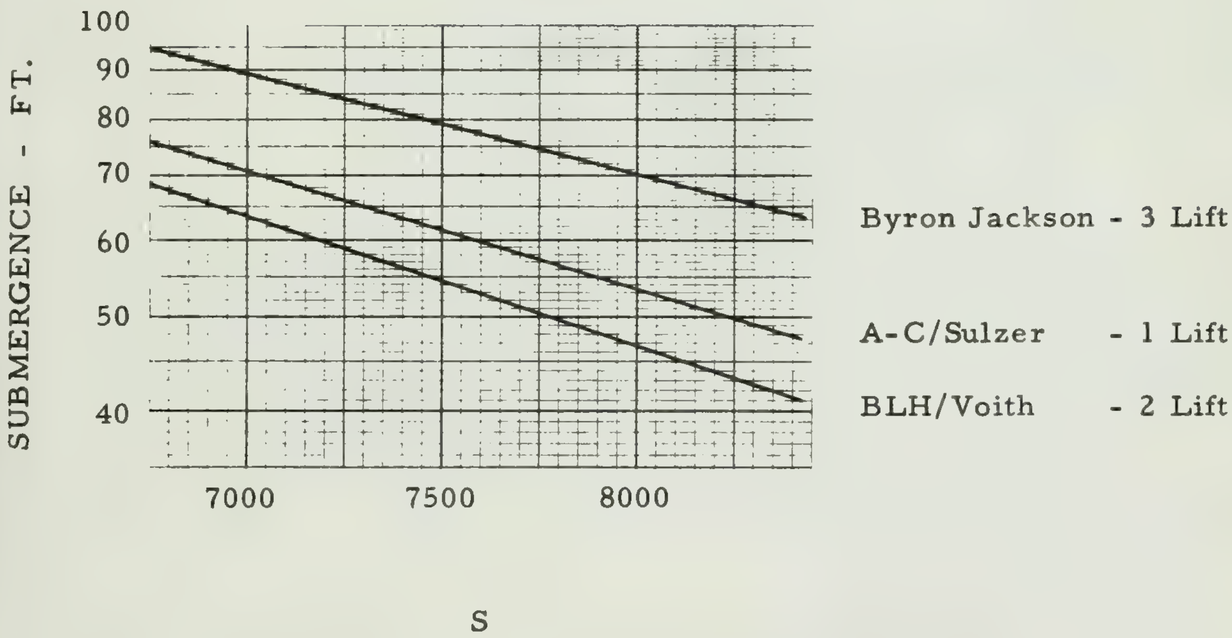
[2] h_v = 1.2 feet at 80°F = vapor pressure (head).

[3] $h_{\ell s}$ is assumed equal to zero.

FIG. 3-9

REQUIRED SUBMERGENCE AT PLANT NO. 1

AS A FUNCTION OF S



5. Cavitation Resistance of Materials

a. Introduction

The resistance of materials to cavitation erosion has been a major subject for research. The mechanism of cavitation damage and its relation to corrosion and other forms of erosion has been, and is being investigated extensively. Generally, stainless steel has better erosion resistance than other materials of construction, but accurate prediction of rates of wear has not been possible. Relative ratings of materials have been made, but even these ratings are somewhat dependent on the kind of cavitation erosion test employed.

Accelerated testing using magnetostrictive oscillators has been popular because of its relative simplicity and fast production of results. Other test devices include cavitating Venturies, rotating disks and specimens in cavitation tunnels. The results of the various investigations are all very interesting but the recommendations for materials is somewhat confusing. The economic considerations are usually not included. Results of some of the many studies are given in the next section.

b. Cavitation Erosion Tests

(1) In a recent paper by Leith and McIlquham[1], comparative test results are presented which also report the relative sand erosion rating. FIG. 3-10 gives their results. It is interesting to note that the cavitation erosion resistance and sand erosion resistance do not correspond. In selecting construction materials, both forms of erosion and corrosion must be considered.

(2) In 1956 the ASME conducted a symposium and issued a report[2] that contained comments on field experience and concluded with the following generalized ratings:

Order of Cavitation Resistance of Metals

Based upon field experience:

1 Stellite
2 17-7 Cr-Ni stainless-steel weld
3 18-8 Cr-Ni stainless-steel weld

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[1] "Accelerated Cavitation Erosion and Sand Erosion", by W. C. Leith and W. S. McIlquham, ASTM Special Technical Publication No. 307, 1961.

[2] Report of 1956 Cavitation Symposium by Wm. J. Rheingons, "Resistance of Various Materials to Cavitation Damage", ASME, 1956.

4 Ampco No. 10 weld
5 25-20 Cr-Ni weld
6 Eutectic-Xyron 2-24 weld
7 Ampco bronze castings
8 18-8 Cr-Ni cast stainless
9 Nickel-aluminum bronze, cast
10 13% Cr cast
11 Manganese bronze, cast
12 18-8 stainless spray metallizing
13 cast steel
14 Bronze
15 Rubber
16 Cast Iron
17 Aluminum

Based upon laboratory tests:

1 Stellite
2 Two layers 17-7 Cr-Ni stainless steel weld
3 18-8 Cr-Ni stainless steel weld
4 Ampco No. 10 weld
5 Cast Ampco No. 18 bronze
6 Nickel-aluminum bronze
7 18-8 Cr-Ni cast stainless
8 13% Cr, cast stainless
9 Manganese bronze, cast
10 Cast steel
11 Bronze
12 Cast iron
13 Sprayed stainless 18-8 Cr-Ni
14 Rubber
15 Aluminum

(3) Results of rotating disk cavitation erosion tests are reported by Lichtman and Weingram[1] which include comparisons with other types of tests and include performance of several newer materials. A table of their results is given in FIG. 3-11.

[1] "The Use of Rotating Disk Apparatus in Determining Cavitation Erosion Resistance of Materials", by J. Z. Lichtman and E. R. Weingram, ASME Symposium on Cavitation Research Facilities and Techniques, 1964.

—COMPARATIVE TESTS—CAVITATION EROSION AND SAND EROSION.

METAL	CONDITION	SPECIFICATION	TENSILE STRENGTH, 1000 PSI	BRINELL HARDNESS	RELATIVE CAVITATION EROSION			RELATIVE SAND EROSION
					VIBRATORY	DROP IMPACT	FLOW	
YELLOW BRASS	ROLLED	B-16	55	140	140	36	59	27
ALUMINUM	ROLLED	2-SO	13.5	21	200	75	47	62
ALUMINUM	ROLLED	65-ST4	35	68	100	69	104	
ALUMINUM	ROLLED	54-SF	43	92	70	56	82	
ALUMINUM	ROLLED	26-ST6	64	156	45	42	78	
ALUMINUM	ROLLED	75-ST6	90	188	31	42	73	22
IRON	CAST	A-48	30	190	108			24
IRON	CAST	A-48	40	200	91	180	315	22
IRON	CAST	A-48	50	220	73			18
RED BRONZE	CAST	B-62	34	65	157			21
ADMIRALTY BRONZE	CAST	B-143	45	90	87			17
PMG BRONZE	CAST	80-20	40	132	41			23
MANGANESE BRONZE	CAST	B-22	100	225	37	33	27	22
ALUMINUM BRONZE	CAST	NIKALIUM	90	170	25			19
ALUMINUM BRONZE	CAST	AMPCO 18	90	180	15	15	15	17
CARBON STEEL	CAST	A-27	70	135	64			11
CARBON STEEL	ROLLED	1020D	75	120	86	7	22	11
CARBON STEEL	ROLLED	1040	75	170	26			10
CARBON STEEL	ROLLED	1040		440	2			7.5
STAINLESS STEEL	CAST	410	75	185	71			9.5
STAINLESS STEEL	CAST	410	75	215	25			8
STAINLESS STEEL	CAST	304	85	140	19	21	14	7.5
STAINLESS STEEL	CAST	202	88	170	22			12
CARBON STEEL	WELD	E 6012		170	42			11
STAINLESS STEEL	WELD	E 301		375	10			7.5
STAINLESS STEEL	WELD	E 308		250	9			9
ALLOY STEEL	WELD	COLMONOY 6		650	6			3
ALLOY STEEL	WELD	STELLITE 6		525	2			2.5
NICKEL	PLATE	HARD Ni			17			11
CHROMIUM	PLATE	HARD Cr			1			1
REFERENCE					LEITH	MATHIESON & HOBBS		STAUFFER

FIG. 3-10 CAVITATION AND SAND EROSION TESTS REPORTED BY LEITH & McILQUHAM

CAVITATION EROSION AND CORROSION PROPERTIES OF TEST MATERIALS

Test Material	NAVAPLSCIENLAB Rotating Disk Cavitation Erosion Rate, µl/hr at 150 fps Fresh Water	Sea Water	Magnetostriction Cavitation Erosion Data In./Yr. [7-9], [16]	Corrosion Fatigue, Rotating Beam Cycles to failure [8], [17]	Static Corrosion Data mpy 12 month continuous [18][3]	Jet Erosion-Corrosion Rate in Sea Water mpy unwelded [7], [9]
nconel 718	0.29	0.28	0.37	$>100(10^5)$ [5]	<0.1 F100	3
(Monel	0.39	0.42	1.06	$>100(10^5)$ [5]	1.1 PC F10-20	9
7-4 PH (H1025)	0.45	0.48	2.03	$>100(10^5)$ [5]	1.8 P F100	16
7-4 PH (1075)	0.29	0.47	1.31	$>100(10^5)$ [5]	1.8 P F100	12
i6A14V	–	0.48	0.87	$>100(10^5)$ [5]	0 F100	0.9 [6]
i8A12Cb1Ta	–	0.51	0.68	$100(10^5)$	0 F100	1.0
erylco 25	–	0.45	1.31	$51\text{-}58(10^5)$	0.5-0.8	50 [6]
D4MCu	–	0.27	1.81	$6.3\text{-}15.3(10^5)$	0.2-0.4 [4]	2 [7]
M 355	–	0.15	1.25	$6.8\text{-}11.8(10^5)$	1.7-2.1 P F100	2.3 [7]
IY-100 (for cladding)	–	–	4.19	–	–	125
ISI 4330M (for cladding)	–	1.1	2.00	$5.0\text{-}5.4(10^5)$ $50.3\text{-}100(10^5)$ [8]	–	325
ISI 4330M (base for coating)	–	0.33	–	–	–	–
astelloy C (for cladding on HY-100)	–	0.48	0.56	–	0.1 F100	2
astelloy C (for cladding on AISI 4330M)	–	0.57	0.44	$100\text{-}125(10^5)$	0.3 F100	4
ild Steel (12 in. dia. disks)	1.70	2.27	–	–	–	–
eoprene	adh.sep. [1] no dam. [2]	–	no weight loss	–	–	–
olyurethane	erosion damage [1] [2]	–	no weight loss	–	–	–

otes: [1] 20 mil thick

[2] 60 mil thick

[3] F = % fouling, P = pitting, C = Corrosion

[4] Specimen cracked

[5] USNEES Reports C3645, 91078B, 910144, 040096

[6] Reference (16)

[7] 10 day exposure

[8] Polyurethane coated 4330M

FIG. 3-11 MATERIAL DAMAGE TESTS REPORTED BY LICHTMAN AND WEINGRAM

c. Discussion

(1) Additional Study

The literature on cavitation damage is quite extensive and yet no one has ever really consolidated it into a comprehensive application study. Since the choice of construction materials may be made at a later date, the study of cavitation erosion and materials will be continued and will include consideration of cost. Also, a modest amount of testing using a magnetostrictive oscillator will be conducted in conjunction with the wear test program. (See Chapter 10).

(2) Tentative Materials Consideration

The test results and comparative ratings given above show that the most resistant materials are of an expensive nature and some are not suited for large complete parts. For example, the Stellite and welded stainless materials are suitable for surfacing and repair work. Other materials must be used for castings. The aluminum bronze and 18-8 (Type 304) or 13% chrome (Type 410) stainless are the most likely candidates for impeller castings. A-C/Sulzer has suggested using cast CA-15 (equiv. to wrought type 410) and Byron Jackson has suggested bronze without being specific about the alloy. With conservative plant designs providing substantial submergence, use of any of the above mentioned materials will probably be satisfactory. But, if submergence is reduced for any reason, then closer attention must be given to materials selection.

CHAPTER 4

MOTOR STUDIES

A. PURPOSE AND SCOPE

The purpose of the study is to examine the motors from the standpoint of overall feasibility and to select motor characteristics which will provide optimum benefits resulting from weighing of all aspects of the motor design including cost, efficiency, reliability, ease of operation and maintenance, as well as load and starting characteristics which could effect savings in the design of the power supply system.

The desired method of operation for the Tehachapi pumps requires motors which will be self-starting when solidly connected to watered pumps and which will be brought up to speed with the pump discharge valve closed. Motors to meet these operational demands have never been built and, therefore, must be examined without the benefit of experience with existing units of equal size and speed and capable of the starting duty desired.

The major purpose of this study is to examine and evaluate the design feasibility of the proposed motors. It is also the purpose of this report to present a comparative evaluation of the motors for a single-lift, two-lift and a three-lift system.

B. EARLY STUDIES AND PRELIMINARY FINDINGS

1. General

During early investigations and as expected, the only United States manufacturers to express desire and indicate a capability to provide the Tehachapi motors were Westinghouse Corporation, General Electric Company and Allis Chalmers. Consequently, these three companies, together with certain motor manufacturers in Europe, have been the major sources of design and cost information.

The use of synchronous motors was established in previous studies because of their lower cost, higher efficiency and better power factor. Also in previous studies, pumps were considered at five different synchronous speeds of 514, 600, 720, 900 and 1200 RPM. The relative merits of horizontal and vertical motors were examined and the use of hydrogen cooling for motors of 720, 900 and 1200 RPM was also considered. In earlier studies, it became apparent that the design problems involved with high peripheral speeds and stresses imposed by heating and magnetic forces on amortisseur windings

and stator coils, during self-starting increased in proportion to the speed required. As a result, the technical and economic feasibility of 720, 900 and 1200 RPM machines was seriously questioned. Nevertheless, the three major United States manufacturers who were conferred with at that time felt that machines with speeds of 600 RPM or less could be provided without encountering serious problems.

2. Motor Starting Problems

Motor starting problems were considered to a limited extent in earlier investigations primarily with respect to 720 through 1200 RPM. As a result, several significant conclusions were reached.

a. Starting of the motors is closely related to the power system supplying the station with respect to allowable starting KVA and the voltage which can be maintained at the motor terminals during initial start, the period of accelerating and the time of pulling into synchronism.

b. Across-the-line starting, because of the high torques required, should be used if possible for motors starting with a watered pump.

c. Reduced voltage starting, because of the corresponding reduction in motor torque must have full voltage applied before pulling into step if the pump is started watered.

d. Reduced voltage starting does not benefit the amortisseur windings.

e. A power system capable of delivering 4,000,000 short circuit KVA to Tehachapi and which will permit a starting inrush of 250,000 KVA should be adequate for across-the-line starting of the largest motors being considered. This was based on the assumption that the selection of transformers and control switchgear would be carefully coordinated with the motor as well as the power system characteristics.

3. Alternate Starting Methods

In previous studies, several motor starting methods were considered as alternates to across-the-line starting in order to alleviate the severe heating and mechanical stresses imposed on the motors during the starting periods. These alternate methods included:

a. Use of starting motors

b. Transformer reduced voltage

c. Reactor reduced voltage

d. Synchronous starting, using a hydro-turbine driven generator or wound rotor motor generator set to start at rest while electrically connected to a pump motor and accelerate to full speed in synchronism with the motor, at which time the pump motor would be transferred to the normal supply.

C. CURRENT CRITERIA

Since the earlier studies, the speeds of the different sized motors have been reduced; 600 RPM for the single-lift and two-lift systems, and 514 RPM for the three-lift system have been established as the only speeds to be considered. Pump arrangements suitable for the use of horizontal motors have been eliminated. It is apparent that the use of hydrogen cooling for motors of 514 and 600 RPM cannot be justified. As a result, the possible choice of motor types and speeds has been materially reduced, which allows the current studies to proceed in a smaller but more precisely defined area of the ultimate pump and motor requirements.

To meet the present requirements, motors for any of the alternate lift systems are to be vertical synchronous type to operate from a nominal 13,800-volt supply. All motors are rated for 40% overspeed, which is the maximum speed reached in the reverse direction, resulting from loss of power or other failure which will cause water to flow through the unit opposite to the pumping direction. Unity power factor motors have been selected to this date, due to cost advantage of the motors as well as the associated electrical equipment. However, the final choice of motors should consider the possibility of benefits to the power supply system which might result from selection of motors rated for leading power factor operation.

Motors are to start with the pump watered and with the discharge valve closed. Starting, accelerating and pull-in torques shall be adequate to take the pumps from standstill to steady-state operating condition.

Additional general criteria, as well as specific criteria for motors which power the single-lift, two-lift and three-lift systems is set forth in the "Basis for Design" at the end of this section. The "Basis for Design" was developed in conjunction with meetings held in Los Angeles between October 27 and December 8, 1964, with representatives from the main offices of Allis Chalmers, General Electric and Westinghouse, and was used in soliciting the motor information from these manufacturers which is listed in Table 4 - I.

The feasibility of designing motors to reliably meet the criteria contained in the "Basis for Design" has been seriously questioned. There is, however, much evidence to indicate that knowledge and experience with very large motors is developing at a very rapid rate and will be attended by a corresponding advancement in the art of motor design and manufacture. For this reason, together with the assurances from Westinghouse and Allis Chalmers, it is assumed that at the time the motors are required at Tehachapi they will be available from competitive bidders fully adequate for the desired full voltage start of the watered pumps.

All data used to evaluate the motors for each of the three alternate lift systems are therefore based on the use of full voltage starting of motors directly connected to watered pumps.

It should not be concluded however, that the feasibility of self-starting motors has been completely established. The starting problems and alternate starting methods have been studied at some length as reported herein, and should continue to be studied until a final firm solution has been reached.

D. MOTOR STARTING REQUIREMENTS

The starting requirements involve the design and control of motors to provide:

a. Adequate means to start, accelerate and pull the watered pump motors into synchronism

b. Ability to withstand a starting frequency of one start per day which will occur during the early life of the station when the plant is operating on off-peak power

c. Characteristics to limit the starting inrush to values within permissible limits of the power system supplying the plant.

In December, 1963, visits were made to the main plants of Allis-Chalmers, Westinghouse and General Electric, and the preceding problems were discussed in detail with their engineers. At that time all three manufacturers indicated a willingness and capability to provide motors of 75,000 to 80,000 horsepower for full voltage across-the-line starting at speeds through 600 RPM which would meet the desired load and operating conditions. As previously stated however, considerable apprehension was expressed, for providing corresponding motors with higher speeds of 720 through 1200 RPM.

When these companies were again contacted in October through December of 1964, Allis Chalmers and Westinghouse reaffirmed their ability to provide motors to meet the design conditions with full voltage starting. General Electric Company, however, felt that the desired full voltage starting duty for the 80,000 and 70,000 horsepower, 600 RPM motors was too severe to enable them to guarantee such motors for an acceptable number of full voltage watered starts. This change from their previous position with respect to 600 RPM motors resulted from design and tests which they have recently made on large machines and subsequent to the first meetings which were held at their main plant in December 1963. They have, therefore, proposed as an alternate that the motors be non-self-starting with the motors to be started and brought up to speed by auxiliary means using a back-to-back synchronous starting method. General Electric did concede, however, that the smaller and slower 46,000 horsepower, 514 RPM motors could be reliably started at full voltage and guaranteed by them for an acceptable number of starts before requiring excessive maintenance. General Electric further stated that they are conducting an exhaustive development program to investigate and seek solutions to the motor starting problems, and at the time that the motors are bid, will probably be able to make more definite recommendations which will possibly revise their present stand.

Although previous investigations had led to the conclusion that motors of 600 RPM and less would not entail critical design problems, the present concern expressed by the General Electric Company has created the need to consider the starting problem more carefully. To further the study, especially with respect to the motor starting, Motor Columbus has conferred with major motor manufacturers in Switzerland and has prepared a report entitled "Swiss Practice in Large High-Speed Synchronous Machines" which is included at the end of this chapter.

1. European vs. U.S. Practice

The Motor Columbus report indicates a strong preference by Swiss manufacturers for the use of solid poles in lieu of laminated poles and amortisseur bars for the application under consideration. They feel, in fact, that if the motors are to be self-started, solid poles are the only solution with respect to design of the rotor. The U.S. manufacturers, on the other hand, when questioned on the use of solid poles were unanimous in preferring the use of laminated poles because of their greater latitude for designing for optimum overall torque characteristics to match breakaway, accelerating and pull-in requirements. They also preferred laminated poles because of the advantage of having smaller eddy current losses in the pole assemblies.

It is concluded in reviewing these opposing viewpoints, that Europeans are far more experienced than Americans in the use of solid pole machines while Americans are more experienced in the use of laminated poles with Amortisseur windings. With due consideration being given to the greater experience of European firms with solid pole machines, it is important to note that according to Swiss claims, the solid pole design will solve all of the amortisseur winding problems, which are considered by all of the American manufacturers to be one of the two most critical areas in designing the Tehachapi motors.

However, it should be noted that from the Motor-Columbus report, the Swiss firms have estimated a required starting inrush of 5.5 to 5.7 times rated current as compared to corresponding values of 4.1 to 4.7 from U.S. companies, which indicates an advantage of the American design to limit the starting surge to a smaller value.

The general assessment of the design of the stator windings with respect to the requirements for special bracing of the coil ends is much the same with the Swiss firms as with the firms contacted in this country. The Swiss firms indicate that excessive maintenance, possibly involving a yearly major overhaul of the stator windings is to be expected unless reduced voltage is used to alleviate the severe mechanical stresses on the coil ends. This attitude supports the feeling of General Electric in this apparently critical area. The attitude of Allis Chalmers and especially Westinghouse, however, is much more optimistic. Westinghouse has very strongly expressed confidence in the ability of their system of insulation and bracing to withstand full voltage starting frequencies of one start per day without any decrease in the life of the motors.

2. <u>Full Voltage Starting</u>

Full voltage starting with watered pumps is the most desirable method of starting because of the low cost of starting equipment, and the simplicity of control and operation. It would also provide the fastest method of getting all units into operation. With this method, the elapsed time to start 16 or more units would probably be limited only by the rate of power increase which could be permitted by the power system supplying the stations. Although full voltage starting is the most desired, it is also the most adverse method from the standpoint of imposing mechanical and thermal stresses on the motors.

The stresses which cause concern include those which result in flexing of the coil ends where they overhang the stator core due to strong magnetic force associated with the high currents in the stator during the

starting period. With laminated pole machines, they also include the very severe duty placed on the amortisseur windings due to the heavy, induced currents when starting as an induction motor. To produce the necessary starting and pull-up torques, the amortisseur bars embedded in the pole faces must be of very appreciable size and also must be continuously connected between poles as opposed to non-continuous connected windings which are adequate for machines operating only as generators, or motors using a non-induction start. The connections between poles must be carefully designed and constructed to withstand the heat absorbed during the starting period and the associated movement of the bars due to thermal expansion and contraction. To appreciate this problem it should be understood that during the starting period, the amortisseur windings must absorb energy equivalent to the energy in the rotating mass of the machine in addition to that required to overcome the load torque. Since copper and copper alloy bars lose strength rapidly with increased temperature, and are vulnerable to fatigue failure with repeated starting, it is important that means be provided for transmitting as much heat as possible into the pole body and that the heat absorbed will not exceed safe limits for the amortisseur winding. It should be noted that the energy to be absorbed by the amortisseur winding is essentially the same for either full voltage starting or reduced voltage starting. For this reason, reduced voltage starting has very little, if any, beneficial effect on the amortisseur windings when using an induction type start.

In exploring the feasibility of motors to reliably provide the desired frequency of full voltage starting, Westinghouse engineers were asked to explain their confidence in overcoming the problems involved. With respect to the stator problems, they place their confidence in their patented "Thermolastic" insulation which they feel is superior to any other system of insulation yet developed, and when used with their established bracing techniques, "Thermolastic" overcomes the problem resulting from flexing of coil ends as well as thermal expansion and contraction. The "Thermolastic" insulation consists of mica tape made with a special bonding resin. This is applied to the coils and wrapped with an outer glass tape and the entire coil impregnated with a patented solventless resin. The coils are then heat treated at their operating temperature to produce an elastic solid insulation. The coil ends, after assembly in the stator core, are spaced with dacron felt between coils and then completely impregnated with a special epoxy resin. End rings are also incorporated in the end bracing and the assembly air-cured at room temperature. This type of construction is used in the 3,600 RPM short circuit generators used in the Westinghouse high voltage test laboratory which is repeatedly subjected to 3,000,000 KVA short circuits, a duty which was described as being far more severe than the starting duty required at Tehachapi.

When questioned about their ability to overcome the amortisseur winding problems, Westinghouse admitted that brazing of bars to the end rings was very critical and largely dependent on the skill of individual workmen. They are

nevertheless confident of their ability to overcome these problems, stating that it is essentially a matter of designing to meet the required duty. They reported that one of the 65,000 HP units at Grand Coulee, which was designed for synchronous starting, was accidentally started at full voltage with the brakes on and yet withstood this extreme starting condition for which it was not designed, with only nominal damage to the damper windings.

Westinghouse has made computer studies which indicate that the required amortisseur bars would be 1-inch x 2-inch copper with 1-1/2 inches between bars. There would be six per pole and they would be deeply embedded in the poles and designed to favor pull-in torque. The pole faces would be approximately 17 inches across and 80-to-100 inches in length with approximately 20-inch spacing of poles around the rotor circumference.

Westinghouse also expressed their feeling that if the system can stand it, full voltage, across-the-line starting is to be preferred and that the motors should be so designed even if they have to be started by alternate methods during the early years of operation due to limitations of the power supply. Neutral reactor reduced voltage starting was suggested as an alternate during the period.

While the problems of full voltage starting should not be minimized, it is logical to assume that the problems (excluding power supply problems) can be overcome by an intensive effort to provide an assured solution. The Motor Columbus report reflects strong confidence on the part of Swiss designers that the amortisseur winding problem can be eliminated with the use of solid poles, and Westinghouse, among American manufacturers, have been very convincing in their claims that their system of insulation and bracing will solve the stator problem (even though they are less convincing in their claims that their system can withstand the full voltage inrush, a combining (if required) of U. S. and European design practices should be able to produce motors to meet the desired requirements.

3. Alternate Starting Methods

Since the configuration and characteristics of the high voltage power supply to the Tehachapi Station have never been firmly established, assumed values have been used to define the stiffness of the supply system and to set values for allowable inrush KVA. These values were established as a result of informal discussions with the Southern California Edison Company in December, 1963. At that time, as a basis for discussing design characteristics with prospective motor suppliers, it was assumed that the power to supply the motors would be delivered by a power system capable of producing a 230 KV short curcuit of 4,000,000 KVA at the Tehachapi site and also permit a motor starting inrush of 250,000 KVA.

The figures of 4,000,000 short circuit KVA and the limit for inrush of 250,000 KVA are assumed values only, with the possibility existing that the system may be only partially developed to its ultimate size. For this reason, and also because of previously mentioned concern with the severe stresses on the stator coils and amortisseur bars, occurring during full voltage starting, a number of alternate starting methods have been examined. The alternates to full voltage starting which have been considered include the following:

a. Reduced voltage with watered pumps

b. Reduced voltage with dewatered pumps

c. Starting motors with dewatered pumps

d. Back-to-back, synchronous starting with watered pumps

e. Back-to-back, reduced frequency starting with watered pumps

f. Unloaded full voltage starting with the motor connected to the pump through a torque converter.

Method a. would alleviate the starting problem with respect to the stress on the stator coils but would provide no appreciable benefit to the amortisseur windings. To provide the necessary pull-in torque, the transfer to full voltage would have to be made prior to pull-in. Open transition starting methods are not recommended due to the possibility of high surges which can result from the open transition during switching from starting to running voltage prior to pull-in. For this reason, the reactor form of reduced voltage starting is recommended. The feeling among the U.S. firms is, that reduced voltage starting should be used in preference to full voltage starting, only if the high values of starting KVA resulting from full voltage starting cannot be tolerated by the power system supplying the plant.

Methods b. and c. have the disadvantage of requiring the pumps to be dewatered, which is undesirable for any of the pumping schemes and particularly so for the use of four-stage pumps used for the single-lift arrangement. Theoretically, method c. could be used with a watered pump but the starting motor would be of such a large size as to prohibit its use.

The back-to-back synchronous starting method d. has been recommended by General Electric Company and is the method for which the 65,000 horsepower pumps at Grand Coulee Dam were designed. While the Grand Coulee pumps were designed for synchronous start, they are presently being operated by the reduced frequency starting method e.

Synchronous starting is accomplished by tying a motor to a generator electrically at standstill, applying field current to both units, and then starting

the generator with either a hydro-turbine or a wound rotor motor being used as the prime mover. The motor is then brought up to rated speed in synchronism with the generator, at which time both units may be synchronized with and connected to the normal power supply. The starting generator may then be taken off the line and be re-used for starting additional motors as required. Separate motor-driven exciters are required for this method as well as means for braking the starting generator to standstill in preparation for starting the next pump. For braking the starting unit, it is proposed to use dynamic braking after the generator is removed from the normal supply bus.

Reduced frequency starting is accomplished similar to synchronous starting except that the two units are electrically connected together with the generator operating at approximately 50 per cent of rated speed. The motor then starts as an induction motor at the reduced frequency. The generator speed decreases as the motor is accelerated until synchronism between the generator and motor is accomplished. The two units in synchronism are then accelerated to rated speed through control of the prime mover at which time the motor is transferred to the normal supply bus by the same procedure used for the synchronous start. This method eliminates the need for dynamic braking of the unit for subsequent starting use. With this method, the starting duty on the stator coil ends and amortisseur windings becomes greater than required for the synchronous start but is nevertheless, far less severe than required for the full voltage induction type start.

Torque converters, method f. have been used in several European pumping installations. By this method, the motor is started and pulled into step unloaded, and then by means of a hydraulic torque converter the pump is pulled up to synchronous speed at which time the driving shaft of the motor is locked to the driven shaft of the pump followed by removal of the hydraulic fluid from the converter.

The use of any one of methods d., e., or f. will greatly alleviate the motor starting problems involved in the design of the motors, but at the expense of high additional cost and increased complexity in the starting operation. As an example, General Electric, in proposing starting method d., provided supplemental information on a wound-rotor driven M-G set which would adequately provide the necessary starting unit. The following is quoted from G. E.'s letter:

"A number of different pump ratings are under discussion. The starting M-G set which we have investigated will be suitable for starting the largest of these, or the 80,000 HP rating. It would be rated as follows:

39,000 KVA Gen.
.95 P.F.
(50,000 HP output)
Estimating price: $1,100,000.

"Approximate dimensions:

Length:	45 ft.
Width :	16 ft.
Height :	14 ft. (including 3 ft. for base)

"The wound rotor motor driving the set will have one less pair of poles in order to be able to synchronize to the bus. The generator and motor rated speed which we have used is 600 and 720, respectively.

"We would expect that the most economical method of feeding this unit is directly from the 13.8 KV bus to which the pump motor is connected.

"A single liquid rheostat can be supplied to handle this size wound rotor motor. Data is as follows:

Rheostat dimensions:	16' x 15' x 17' high
Two heat exchangers :	24' long x 40" in diameter
Estimated price (incl. amplidyne control:	$ 185,999.

"We would plan to use dynamic braking to bring the M-G set to a stop for starting the next pump. This would be done by applying a DC voltage to the wound rotor motor primary after it has been tripped off the bus. We estimate that a 125-volt, 125 KW M-G set would be suitable for this purpose. The cost of this item is negligible with respect to the other items; however, a separate metalclad breaker will be required to connect this small set to the W. R. motor primary.

"The approximate accelerating time would be one minute. Assuming a minute for starting auxiliaries and another minute for synchronizing, the total starting time would be of the order of three minutes.

"We have also taken a look at reduced frequency induction starting and it does not appear that this has any advantage over the synchronous start other than the fact that no dynamic braking is needed. The attached curve illustrates this type of start. The available induction motor torque is small, so synchronizing takes place at a low motor speed. M-G exciters would be required for either type start so there would not be any significant difference in estimated costs for the two schemes.

"We believe that the synchronous start would be the simplest from the control standpoint."

If adequate reliability is to be provided, the above starting units would have to be provided in duplicate to insure that one unit would always be available for starting use. A further additional cost would also be required for additional bus work and switching equipment.

Because of the numerous disadvantages associated with the alternates studied, full voltage starting has been made a part of the current criteria for motors used in all of the lift schemes considered in this report. The feasibility of using full voltage starting has been supported by replies from two of the three manufacturers to which inquiries were directed. As previously pointed out, however, this criteria should be accepted with some reservations, and the final design criteria for the motors under consideration should be culminated only after a more thorough detailed study has been made of the motors in conjunction with the power system supplying the plant. This study should encompass the power supply from the source(s) of generated power to the point where power is delivered to the motor terminals, and after all of the pump requirements have been established in their final form. Such a study should make use of an AC network analyzer or involve the use of a suitable computer. The study should consider the effects of motor starting surges on motor performance and the attendant effects on the power system. It should also include load flow studies under normal and emergency conditions, establish values of fault duties and system stability limits if they are adversely affected by changing load conditions. While assisting in power system planning, it would also examine the possible need for power factor control and the possible benefits from incorporating a predetermined synchronous condenser capacity into the motors.

4. Summary

The following is a summary of the information developed during numerous meetings with the three major U.S. manufacturers, the study of current technical papers, and a review of the findings set forth in the Motor Columbus report. The Motor Columbus report is included herein at the end of this chapter.

a. The opinion is unanimous among U.S. manufacturers as well as Swiss manufacturers that no particular difficulties will be encountered if self-starting of the motors is not required.

b. The only two critical design areas with self-starting motors are:

(1) The rotor with respect to designing for adequate torque as well as the ability of the amortisseur winding to withstand the severe heating effect created by induced currents while starting as an induction motor.

(2) The stator coils, especially with respect to insulating and bracing of the coil ends to prevent failure due to mechanical flexing resulting from the strong magnetic forces created during full voltage starting.

c. There is a major difference of opinion between U.S. and Swiss manufacturers. Swiss manufacturers feel that rotors with solid poles and without amortisseur windings will, without any great difficulty, eliminate the rotor problems, and is the only reliable solution in designing the rotor. All U. S. manufacturers favor the use of laminated poles and amortisseur bars.

d. There is a wide divergence of opinion between the U.S. firms of Westinghouse and Allis Chalmers as opposed to the General Electric Company with respect to the feasibility of self-starting motors. Westinghouse and Allis Chalmers are on record that self-starting motors for all the sizes and speeds required can be manufactured to reliably start once a day. General Electric feels that only the 46,000 HP, 514 RPM motors can be guaranteed for the required frequency of starting. However, there is no precedent even for the 46,000 HP motor.

e. Full voltage starting as the preferred method, in addition to imposing severe stresses on the motors will also require a power system sufficiently stiff to maintain acceptable voltage levels associated with the high starting inrush requirements.

f. Alternates to full voltage starting involve the following considerations:

(1) Motors started by auxiliary means (non-self-starting) will result in high costs for auxiliary equipment, increased complexity of control, and will require a longer time for start-up.

(2) Reduced voltage starting will not appreciably benefit the rotor but will reduce stresses on the stator coil ends. It can also be used to greatly reduce the magnitude of starting KVA drawn from the power system.

5. Conclusions

a. Self-starting motors are much to be preferred over the use of motors started by auxiliary means.

b. The amortisseur windings present the most critical of all the design problems, if the motors use laminated poles and are to be self-starting with either full voltage or reduced voltage.

c. Reduced voltage starting can be used to appreciably reduce the starting inrush as well as reduce the mechanical stresses on the coil ends of the stator winding, but will not benefit the amortisseur windings to any appreciable extent.

d. The magnitude of the motor starting problems are considered equally critical for both the 80,000 and 70,000 horsepower, 600 RPM motors.

The problems involved with starting the smaller and slower 46,000 horsepower, 514 RPM motors, however, are considerably less than for the larger machines.

e. If we exclude the advantages of starting the 46,000 horsepower motors, the relative advantages of the single-lift, two-lift and three-lift systems for the motors alone are inconclusive because of the wide differences which may result with respect to the cost of the associated electrical equipment. A realistic evaluation must consider the relative advantages of the alternate electrical installations in their entirety.

f. While there is much to support the feeling that reliable self-starting motors can be provided, the apprehensions expressed by the U. S. General Electric Company and certain manufacturers in Switzerland leave the feasibility of these motors open to doubt. Continued effort should be exerted to eliminate this doubt so that the feasibility of the required motors can be accepted without question.

g. In order to examine the motors in greater depth, it is highly important that the characteristics and limits of the power system supplying the motors be established.

TEHACHAPI PUMP MOTORS

BASIS FOR DESIGN

A. DESIGN RATING

Vertical, synchronous, 13,800 volt, 60 cycle, 3 phase, 1.0 P.F. direct connected exciter, 60 C. rise, designed for 40% overspeed.

COMMON FEATURES

Amortisseur windings for across-the-line starting of watered pump with closed discharge. Motor enclosed in air housing with water cooling and provided with high oil pressure lift on thrust bearings.

The following should be included with each motor at the quoted price:

Exciter including exciter cubicle

Air coolers

Air housing, platform, handrails and stairway

Thrust and guide bearings

High pressure lubricating system

Foundation bolts, half coupling and CO_2 system piping

Brakes and jacks

CT's for differential relaying

Space heaters

Heat detectors

Handling equipment and unit accessories

Rail freight to destination

Combination jacks and brakes shall be included which will elevate the rotor and rated to brake 1% of rated torque from 50% speed to standstill in 7-1/2 minutes.

ASSUMPTIONS

General

Motors will be started across-the-line with pumps watered with closed discharge. Starting frequency will equal one start per day.

Power will be supplied from a 230 or 500 KV system capable of delivering a minimum of 4,000,000 short circuit KVA to the high side of the transformers supplying the pump motors. Voltage drop on the high side of these transformers, resulting from starting of individual motors, shall not exceed 5%.

Motors are expected to operate in an ambient of 104 F. maximum with a cooling water temperature of 87 F.

Case 1 (Single Lift)

Units required (for 4-stage pumps)	16
Horsepower each unit	80,000
Speed (RPM)	600
Elevation above sea level (ft.)	1,232
Pump WK^2 (lb. ft.^{2})	190,000
Pump Load on thrust bearing (lbs.)	200,000
Pump Load at full speed with closed discharge (% of motor rated HP)	62

Motors will be connected to one (1) common 250,000 KVA transformer for each four (4) motors.

Case 2 (Two Equal Lift Stations)

Units required (for 2-stage pumps)

Station A	9
Station B	9

Horsepower each unit	70,000

Speed (RPM)	600

Elevation above sea level (ft.)

Station A	1,222
Station B	2,187

Pump WK^2 (lb. ft.^{2})	230,000
Pump Load on thrust bearing	100,000
Pump Load at full speed with closed discharge (% of motor rated HP)	55

Motors will be connected to one (1) common 165,000 KVA transformer for each three (3) motors.

<u>Case 3 (Three Equal Lift Stations)</u>

Units required (for 1-stage pumps)

Station A	9
Station B	9
Station C	9

Horsepower each unit	46,000
Speed (RPM)	514

Elevation above sea level (ft.)

Station A	1,206
Station B	1,850
Station C	2,494

Pump WK^2 (lb. ft.^{2})	80,000
Pump Load on thurst bearing	100,000
Pump Load at full speed with closed discharge (% of motor rated HP)	51

Motors will be connected to one (1) common 110,000 KVA transformer for each three (3) motors.

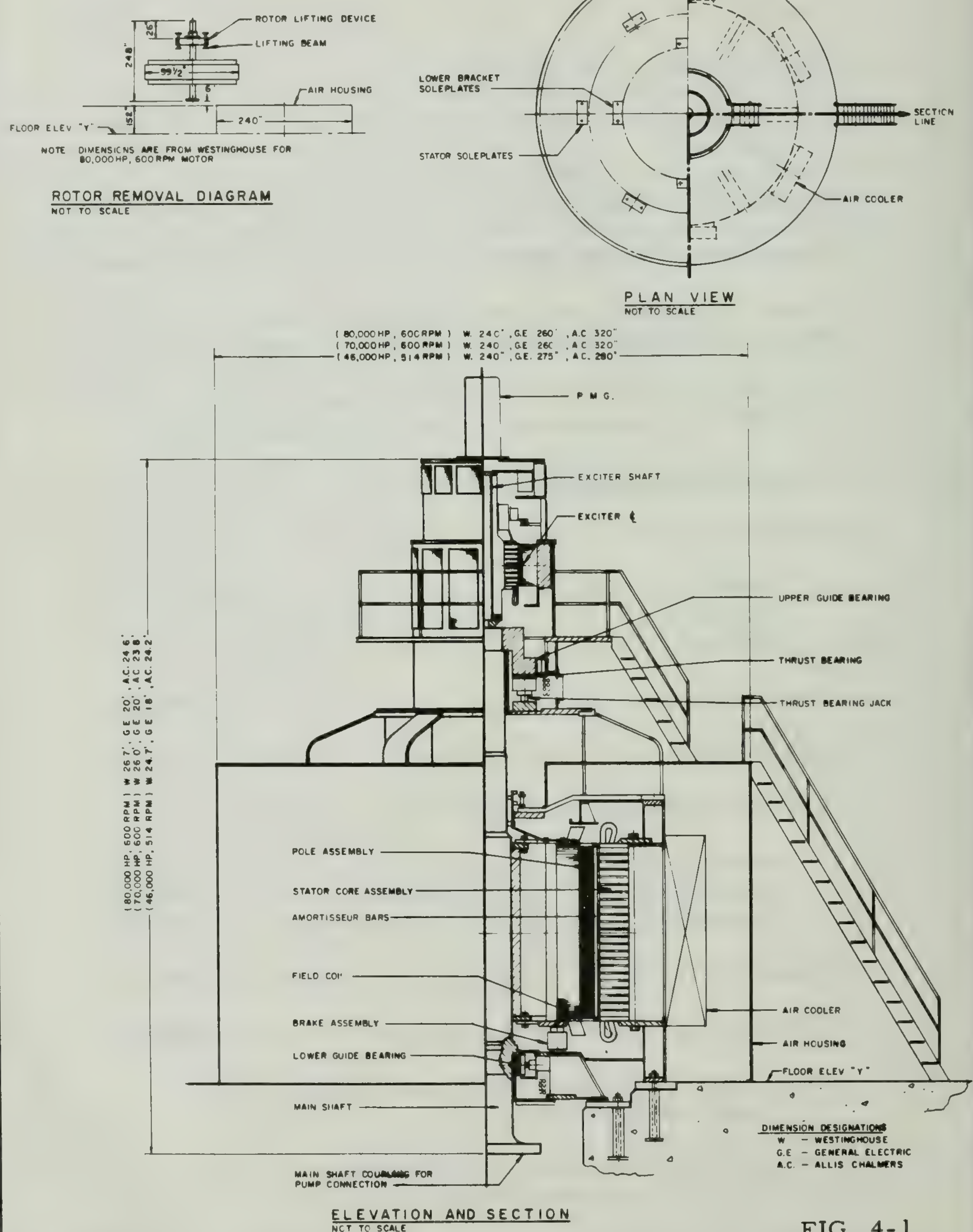


FIG. 4-1

DMJM DANIEL, MANN, JOHNSON, & MENDENHALL 3325 WILSHIRE BLVD. · LOS ANGELES 5, CALIFORNIA · DUNKIRK 1-3663 PLANNING • ARCHITECTURE • ENGINEERING • SYSTEMS	TEHACHAPI PUMPING PLANT MOTOR ARRANGEMENT SINGLE-LIFT (80,000HP, 600RPM) TWO-LIFT (70,000HP, 600RPM) THREE-LIFT (46,000HP, 514RPM)	R BRINK W.E.G.	637-1-1 E-1

TABLE 4-1 MOTOR DATA RECEIVED FROM MOTOR MANUFACTURERS

	Single Lift			Two Lift			Three Lift		
	West.	G. E.	A. C.	West.	G. E.	A. C.	West.	G. E.	A. C.
Horsepower	80,000	*80,000	80,000	70,000	*70,000	70,000	46,000	46,000	46,000
Speed (RPM)	600	600	600	600	600	600	514	514	514
Number of Units	16	16	16	18	18	18	27	27	27
Cost per Unit (Motors only)	$ 880,000	$ 590,000	$ 600,000	$ 704,000	$ 530,000	$ 552,778	$ 496,000	$ 460,000	$ 398,150
Total Cost (Motors only)	14,080,000	9,440,000	9,600,000	12,672,000	9,540,000	9,950,000	13,392,000	12,420,000	10,750,000
Estimated Efficiency including Excitation and Bearing Losses (%)	98.5	97.95	97.90	98.45	98.00	97.80	98.4	97.75	97.50
Inertia, Dimensions and Weights									
Motor WK^2 (Lb. Ft^2)	1,110,000	2,000,000	2,700,000	980,000	1,700,000	2,300,000	750,000	1,500,000	1,380,000
Outside Diameter over Air Housing (In.)	240	260	320	240	260	320	240	275	280
Overall Height including Exciter (Ft)	26.667	20	24.6	26.00	20	23.8	24.7	18	24.2
Rotor Diameter (In.)	99.50	101	136	99.5	101	136	101	110	118
Stator Diameter (In.)	160	157	220	160	157	220	160	170	183
Rotor Weight (Lb.)	172,000	231,000	230,000	157,000	220,000	195,000	135,000	146,000	185,000
Total Weight (Lb.)	410,000	476,000	470,000	380,000	428,000	410,000	385,000	295,000	335,000
Design Characteristics									
Synchronous Reactance Xd (%)	140	92	100	150	92	100	150	92	100
Transient Reactance X'd (%)	29	38	31	31	38	29	31	38	33
Subtransient Reactance X''d (%)	25	25	27	27	25	26	27	25	28
Motor Starting Impedance (%)	25.0	25	27	27	25	26	27	25	28
Motor Pull-In Impedance (%)	51.3	--	50	55	--	50	55	--	50
Motor Starting Power Factor (%)/Motor Pull-In P F (%)	7/38	--	23/53	7/38	--	23/53	7/38	--	25/53
Estimated Speed at Pull-In (% of Syn. Speed)	97	--	95	97	--	95	97	--	95
Estimated Starting Time, Standstill to Pull-In (Secs.)	30	--	15-23	30	--	15-23	30	--	15-23
Locked Rotor KVA at Rated Full Voltage (%)	400	--	370	370	--	385	370	--	350
Starting Torque at Rated Full Voltage (%)	20	--	40	20	--	40	20	--	40
Pull-In Torque at Rated Full Voltage (%)	78	--	100	70	--	100	65	--	100
Pull-In Torque at Rated Full Voltage (%)	150	150	150	150	150	150	150	150	150
Performance Data Associated with Assumed Power Supply									
KW Equivalent of Motor Rated Horsepower	60,600	60,900	61,000	53,100	53,300	53,400	35,100	35,100	35,300
Assumed Impedance of Power System (% on Motor KVA Base)	1.5	1.5	1.5	1.3	1.3	1.3	0.9	0.9	0.9
Assumed Impedance of Transformer Supplying Motor(" " " " ")	3.6	--	2.9	4.7	--	3.9	4.7	4.0	3.9
Motor Starting Inrush (KVA)	201,000	--	195,000	161,000	--	171,000	108,000	117,000	108,000
Estimated Terminal Voltage at Start (%)	81.5	82	80-90	81.7	84	80-90	82	83.7	80-90
Estimated Terminal Voltage at Pull-In(%)	91.5	--	80-90	91.5	--	80-90	92	--	80-90
Starting Torque at Estimated Terminal Voltage (%)	13	--	--	13	--	--	14	--	--
Pull - In Torque at Estimated Terminal Voltage (%)	65	--	--	58	--	--	54	--	--

*MOTORS NOT RECOMMENDED FOR SELF-STARTING BUT BY SYNCHRONOUS BACK-TO-BACK STARTING.

State of California
Department of Water Resources
Sacramento/California

California State Water Plan
Tehachapi Pumping Plant
Motor Study

SWISS PRACTICE

in

LARGE HIGH-SPEED SYNCHRONOUS MACHINES

February 1965

M O T O R - C O L U M B U S
Baden/Switzerland

S u m m a r y

This report was compiled on the request of the Department of Water Resources, to contribute from Swiss practice and experience to the problem of large, high-speed self-starting synchronous motors as anticipated for the Tehachapi pumping plant.

It contains minutes of the meetings held with Swiss motor manufacturers and a summarizing discussion of the information they provided orally and in writing. It deals with the problems of the motor itself as well as the influence of transformer and switching arrangement and network conditions.

Written statements of Swiss motor manufacturers are attached together with translations.

To achieve an overall point of view, alternative starting methods are listed and their possibilities discussed.

The report is closed with conclusions and recommendations.

1. INTRODUCTION

The motors to drive the pumps of the Tehachapi pumping plant will have the following rating:

Single lift	concept, alternative	I	80,000 HP	600 rpm
Two-lift	concept, "	II	70,000 HP	600 rpm
Three-lift	concept, "	III	46,000 HP	514 rpm

Motors of even higher speeds, 900 and 1200 rpm, had been considered during earlier stages of the project, but were abandoned because of objections of the US manufacturers against such high speeds. However, with further investigations, it became evident that, depending on the specific design, difficulties are involved also with 600 and 514 rpm, and at least one US firm expressed serious concern regarding frequent starting of the motors.

On the other hand, Brown Boveri & Cie., one of the leading Swiss firms, had expressed in previous discussions (with Mr. Troost of DWR in August 1963, and with Mr. H. Gartmann of DMJM on August 20, 1964, see memorandum dated October 29, 1964), that they were able to design self-starting motors of this range, and even with 900 and 1200 rpm. This is due to the special rotor design they use for high-speed synchronous machines in general,which is especially suitable for self-starting.

In October 1964, the Department asked Motor-Columbus to acquire more detailed information from Swiss manufacturers on their design methods and recommendations on self-starting synchronous machines of this size.

2. MEETINGS WITH MANUFACTURERS

There are three manufacturers in Switzerland, who are capable to build such motors:

Brown Boveri & Cie.,	Baden	(BBC)
Maschinenfabrik Oerlikon,	Zurich	(MFO)
SA des Ateliers de Sécheron,	Geneva	(SAAS)

Previous to discussions, the manufacturers were informed of the rating of the machines in question, design and operating criteria, and network conditions, by transmitting to them copies of the "Basis for Design" which was compiled by DMJM for a parallel inquiry with US manufacturers. Their statements were requested on the following questions:

a) What would be the general design features of such machine ?

b) Could the motors be built for safe and reliable operation for starting with the pump filled with water ?

c) What are the specific advantages of the recommended design with respect to the given operating conditions ?

d) Could the motor be started across the line, under full voltage, provided the grid would be strong enough ?

e) How would across-the-line starting effect the life-time of the stator winding ?

f) Which system is recommended for reduced voltage starting under the given network conditions ?

2.1 Meeting with Brown Boveri (BBC)

The meeting was held on December 14, 1964, at the Brown Boveri plant Birrfeld near Baden, Switzerland.

BBC:

R. Noser	-	Chief Engineer	Synchronous Machine Department
E. Meyer	-	Head Design Office	
K. Baltisberger	-	Design Engineer	

MC:

G. Leupin	-	Mechanical Dept.,	Rotating Machinery Group
A. Weber	-	Electrical Dept.,	Transformer Group
O. Hartmann	-	Project Engineer	

BBC was somewhat familiar with the Tehachapi project and the motor problems involved from previous discussions. They had also contacted their New York sales office in this matter. Their impression was that the Tehachapi motors would be purchased in the US and that there would be no chance for any foreign manufacturer eventually to be awarded a contract for these machines. For this reason, they did not want to prepare a written technical study, but they were cooperative to discuss the technical problems with us and to give us the desired information orally. Major points of discussion were as follows:

2.1.1 Motor design

BBC said that a synchronous machine with rating 60 MVA (corresponding to 80,000 HP, p.f. 1.0) and 600 rpm would not encounter serious design problems. Mechanical stresses are even less for the Tehachapi motors than would be for hydro-generators of same rating because of the low run-away speed of only 140 % compared with 180 - 200 % run-away speed of hydraulic turbines. Therefore, for normal operation (i.e. not self-starting) the machine could be built with either laminated or solid poles. BBC would prefer solid poles in any case, because this is their standard design for high-speed machines.

If self-starting is required, BBC believes that solid poles are the only solution. Because of limited experience with laminated pole machines they could not say that self-starting would be impossible with laminated poles and amortisseur winding, but they were very doubtful whether the amortisseur

winding could be designed to withstand the thermal and mechanical stresses of repeated starting. They recommend solid poles without any additional interconnections from pole to pole. The pull-in torque of such machine would be lower than with amortisseur winding or with solid poles and interconnections from pole to pole, but still high enough for synchronization under 60 % reactive pump torque.

The motor could be designed for starting under full voltage. However, this would lead to approx. 9 % voltage drop on the high voltage side of the transformer under the given network conditions. 6.5 % voltage drop would be possible with an enlarged motor with higher reactance. With full voltage starting the inrush current would be 5.5 - 5.7 times the rated current, and the stator winding, especially the end turns, would be subject to heavy mechanical stresses. This would require, in BBC's opinion, a yearly major overhaul to check and refix, if necessary, the end turn bracings, even with best insulation and special bracing.

BBC believes, therefore, that reduced voltage starting would be advisable to limit both the voltage drop on the grid and the shock on the stator winding. Voltage reduction of about one-half would be sufficient.

2.1.2 Transformer and switching arrangement

BBC recommends that three or four motors are connected to one transformer as presently anticipated. This arrangement has the advantage that the voltage drop through the transformer is only one-third or one-fourth of that of one transformer per unit, when starting the first motor. When starting the further units connected to the transformer, the units already in operation could support the voltage stability by over-excitation (most of the current drawn during starting is wattless current). With this arrangement, it would be possible to use a transformer of normal impedance.

BBC believes that for reduced voltage starting an impedance coil would be the best solution. Possible switching arrangements of the impedance coil were discussed (see exhibit 3, Fig. 1a, b, c).

2.2 Meeting with Maschinenfabrik Oerlikon (MFO)

The meeting was held on December 17, 1964, at the offices of MFO in Zurich-Oerlikon.

MFO:

F. Seefeld	-	Chief Design Engineer, Synchronous Machines Dept.
Dr. H. Roose	-	Sales Engineer , Export Dept.

MC:

G. Leupin	-	Mechanical Dept.,	Rotating Machinery Group
A. Weber	-	Electrical Dept.,	Transformer Group
O. Hartmann	-	Project Engineer	

MFO was also reluctant to undergo detailed studies on the Tehachapi motors. However, after explaining the situation to them, MFO was prepared to give their recommendations in writing. Mr. F. Seefeld made a very exhaustive write-up, which also covers the discussion we had with them. This write-up is attached to this report (exhibit 4), together with the translation (exhibit 5).

2.3 SA des Ateliers de Sécheron, Geneva (SAAS)

From this firm we received a written statement dated January 13, 1965, which is attached to this report together with the translation (exhibits 6 and 7). Since this statement indicated that their opinions widely coincide with the information received from the two other Swiss firms, a meeting with SAAS was considered dispensable.

3. DISCUSSION OF MANUFACTURERS' INFORMATION

Before going into technical details, some general remarks regarding the three Swiss firms may be of interest.

All three firms were founded late in the last century, and Motor-Columbus' business connections with them date back to this time also. Their manufacturing program covers a wide range of electrical equipment. Due to the geographic conditions in Switzerland with many high-head hydroelectric power developments, one of their activities is the manufacture of high-speed synchronous machines. Many of the hydro-generators they have built in the last 20 years are in the same range of power and speed as the Tehachapi motors. Exhibits 1 and 2 document the experience of the Swiss firms in this field, and it may also be mentioned that with 9 of the 36 machines listed, Motor-Columbus was the consulting engineer on behalf of the power plant owner.

3.1 Motor Design

General design problems encountered with high-speed machines are the following:

a) Mechanical stresses in the rotor, particularly regarding the pole fixation due to the centrifugal forces. This problem is more significant with hydro-generators than with pump motors, because of the higher run-away speeds of hydro-turbines. It is standard in Switzerland to test the complete rotor at run-away speed during 2 minutes in the workshop.

b) Bearings, especially the thrust bearings, critical speed, and vibrations.

c) Thermal dilatations of both pole and stator windings require particular attention with increasing length of the machine. The pole windings have to withstand both mechanical and thermal stresses. These problems become more difficult with frequent stopping and starting of the machine.

While the design features of the stator do not differ basically from those for a low-speed machine, the configuration of the rotor is different from that of a low-speed chain rotor. The rotor consists of a cast steel spider with shrunk -on rolled steel rings, on which the poles are fixed, mainly by dove-tail or similar fixation. With higher speeds and smaller diameters, the rings may be shrunk on the shaft (without spider), or the poles are fixed directly on a central body which may be in one piece with the shaft or with bolted-on half-shafts. This will give an idea of the mostly used designs, but there is a variety of combinations.

Solid cast or forged steel poles are used as well as laminated poles with damper (amortisseur) winding. Two firms (BBC and SAAS) prefer solid poles as standard design, while the third firm (MFO) prefers laminated poles in general. Beside the historical background of the firms' design methods, the respective features of the two pole designs can be characterized as follows:

Solid poles allow higher mechanical stresses. The pole shoes themselves act as a damper winding with high ohmic resistance; a special damper winding can, therefore, be omitted, either for stability of a generator or for starting of a motor. For the latter case, it is essential that the heat developing during the start-up is spread over the mass of the poles, which is much bigger than that of a corresponding amortisseur winding.

Solid poles have higher pole shoe losses due to eddy currents than laminated poles. However, this difference decreases with increasing length of the machine.

Due to these characteristics, also MFO uses solid poles in cases where high mechanical stresses and/or severe starting conditions are prevailing. The latter is the case with the Tehachapi motors and leads to a remarkable coincidence of the design recommendations of all three firms:

A synchronous motor rated 80,000 HP, 600 rpm, 140 % run-away speed, would not present special problems except starting. Mechanical stresses are reasonable due to the low run-away speed. Both laminated and solid poles would be applicable in view of the mechanical stresses.

Self-starting involves two major problems: Inrush shocks on the stator winding and rotor heating.

Mechanical shocks on the stator winding require special bracing of the end turns. Repeated full voltage starting, with inrush current approx. 5.5 rated current, would be possible with respect to the motor, but would require more frequent servicing and refixing of the winding. As the shocks are proportional to the square of the inrush current, they can be limited considerably by reduced voltage starting. Starting with about one-half rated voltage is recommended, which would reduce the inrush current to approx. 2.5 rated current.

The shock problem is the same whether the pump is filled with water or dewatered for starting.

Over-temperatures of the stator winding due to starting are very low and do not raise problems.

The heat developed in the rotor is approx. equal to the mechanical energy developed during the starting process, consisting of the energy necessary to accelerate the unit and the energy dissipated by the pump. This amount of heat would lead to extreme temperatures in an amortisseur winding. With dewatered pump, the starting energy of the pump is decreased to a negligible value, but still the amortisseur winding would reach excessive temperatures.

All three firms stated that they would design the motor with solid poles, to avoid the amortisseur winding. Moreover, MFO calculated the temperatures of the amortisseur winding and came to the express conclusion that a design with amortisseur winding would not be feasible for either one of the three alternatives, even with dewatered pump. According to their calculation, the over-temperatures of the amortisseur winding would reach 300 - 400 °C = 540 - 720 °F, depending on motor alternative.

Such temperatures are deemed to be impermissible as the strength properties of the amortisseur winding material and its connections would be decreased. It is essential that the amortisseur winding has also to withstand high mechanical stresses and thermal stresses due to uneven heating and elongation of the individual rods.

Under the same starting conditions the over-temperature of the solid poles is approx. 130 °C = 235 °F, which does not encounter any problems. Moreover, the heat originated in a relatively thin surface layer can easily dissipate over the whole steel pole body.

The problem of rotor heating is basically the same, whether full voltage or reduced voltage starting is applied.

MFO and SAAS recommend low-resistance connections from pole to pole, while BBC would prefer to avoid even this additional element. The latter is certainly desirable from the reliability standpoint, but reduces the pull-in torque of the machine. This solution would be more sensitive to under-voltage in the network and to voltage drop in the transformer.

3.2 Transformer and Switching Arrangement

As already mentioned reduced voltage starting is not absolutely necessary for the motor, but desirable for the benefit of the stator winding. Reduced voltage starting is also necessary in order to limit the voltage drop in the network to 5 % with the arrangement as anticipated, i.e. three or four units connected to one common transformer.

The voltage reduction can be realized by impedance coils. The starting voltage would be about one-half the full voltage. When the unit has reached approx. 60 % speed, the coil is short-circuited. Exhibit 3 shows three different arrangements of such impedance coils. Economic considerations will be prevailing in the choice of arrangement, taking into account equipment cost, space requirements, and building cost..

MFO recommends full voltage starting, however, with one transformer per unit. Thus, the full transformer impedance (against one-third or one-fourth with three or four units per transformer) would be active and reduce the voltage on the motor terminals during starting and also the inrush current to approx. 3.3 times rated current. Accordingly the voltage drop on the network would be within the limit of 5 %. The principal arrangement is also shown on exhibit 3, figure 2. The simplicity of this solution is obvious, and it would also simplify the unit protection and the automatic control. Space requirements would be less as the 13.8 kV busbar and switchgear are omitted, and failure of one transformer would inflict only one pumping unit. On the other hand, the cost of transformers would probably be higher, and the starting voltage cannot be chosen as freely as with impedance coils.

4. ALTERNATE STARTING METHODS

Full voltage asynchronous starting of the pump motor, with the pump filled with water, is the starting method presently anticipated. There are, however, serious doubts whether this method is technically feasible with motors having an amortisseur winding. Therefore, a brief discussion of alternate methods is of interest.

Exhibit 8 gives a review of possible starting methods. The respective advantages and disadvantages are listed therein, and a few summarizing remarks are mentioned below.

4.1 Dewatered Pump

Besides the disadvantages on the pump itself, it is doubtful whether across-the-line starting of the motor would be possible. Problems with heating of amortisseur winding are reduced, but not eliminated, and the inrush conditions (shock on stator winding and voltage drop on the network) are the same with filled or dewatered pump.

Starting with dewatered pump makes other starting methods technically and economically feasible (pony motor, starting turbine, synchronous starting with starter unit) due to the

low power requirements.

4.2 Torque Converter

With the use of a hydraulic torque converter the filled pump is coupled to the motor at full speed. For the starting of the motor the conditions are the same as if the pump were dewatered and the remarks under 4.1 are true also for this method.

4.3 Pony Motor and Starting Turbine

It is believed, that both methods are feasible only with dewatered pump starting.

4.4 Synchronous (Back to Back) Starting

This method has been used where turbine-generator sets and pump-motor units are installed in the same plant. For Tehachapi the hydraulic or electric starter unit, with necessary spare, would be additional equipment only for starting purpose. Together with the additional electrical equipment the cost would run very high.

Furthermore, this method is very time-consuming. After each start the starter unit has to be stopped, and all the electrical switch-over operations have to be made before the next unit can be started. It would take hours to get 16 units in operation.

5. CONCLUSIONS AND RECOMMENDATIONS

Information received from Swiss motor manufacturers regarding design of the motors for the Tehachapi pumps, together with a review on alternate starting methods lead to the following conclusions:

a) The design of such motors does not encounter particular difficulties, if self-starting is not required.

b) The motors can also be designed for asynchronous self-starting. For this duty a rotor design with solid poles, without amortisseur winding is deemed to be the only reliable solution. Heating of the amortisseur winding would be excessive for either motor alternative, and even with dewatered pump starting.

c) Reduced voltage starting is required if the voltage drop on the network shall be limited to 5 %, but is also advisable to reduce inrush shocks on the stator winding. Impedance coils in various arrangements are recommended. An alternate solution would be "block" arrangement of one transformer for each unit, using the voltage drop in the transformer as to reduce the voltage for starting.

d) Other starting methods than asynchronous self-starting of the pump motor are feasible, but at the expense of considerable extra cost, space requirement, and complication. The many additional elements would reduce the unit and plant reliability.

Based on these considerations,we would like to propose the following recommendations:

a) The feasibility of synchronous motors, suitable for asynchronous self-starting, should be investigated thoroughly and a definite decision should be reached as soon as possible, because it concerns the whole plant layout.

b) Alternate starting methods should be investigated further only, if self-starting motors should be found definitely unfeasible for one or the other reason.

c) Every effort should be made to realize a safe and reliable solution with self-starting motors. US firms should be requested to design and calculate the motors in detail. Only this would allow a clear and definite judgement. Solid pole rotor design should be taken into consideration to avoid the amortisseur winding and the problems encountered herewith.

d) System arrangements for reduced voltage starting should be investigated to achieve an optimum solution.

e) It would be helpful if these investigations could be based on a final lift concept and on more information on the network configuration.

M O T O R - C O L U M B U S
Electrical Management Company Ltd.

w. Kegg — ppa Brunner

<u>Encls.:</u>

see page 16

February 16, 1965
S-Hm/2-Lp/bch

Enclosures

Exhibit 1 : List of large high-speed synchronous machines of Swiss manufacture

Exhibit 2 : Diagram of large high-speed synchronous machines of Swiss manufacture

Exhibit 3 : Possible system arrangements

Exhibit 4 : Letter and write-up from MFO

Exhibit 5 : Translation thereof

Exhibit 6 : Letter of SAAS

Exhibit 7 : Translation thereof

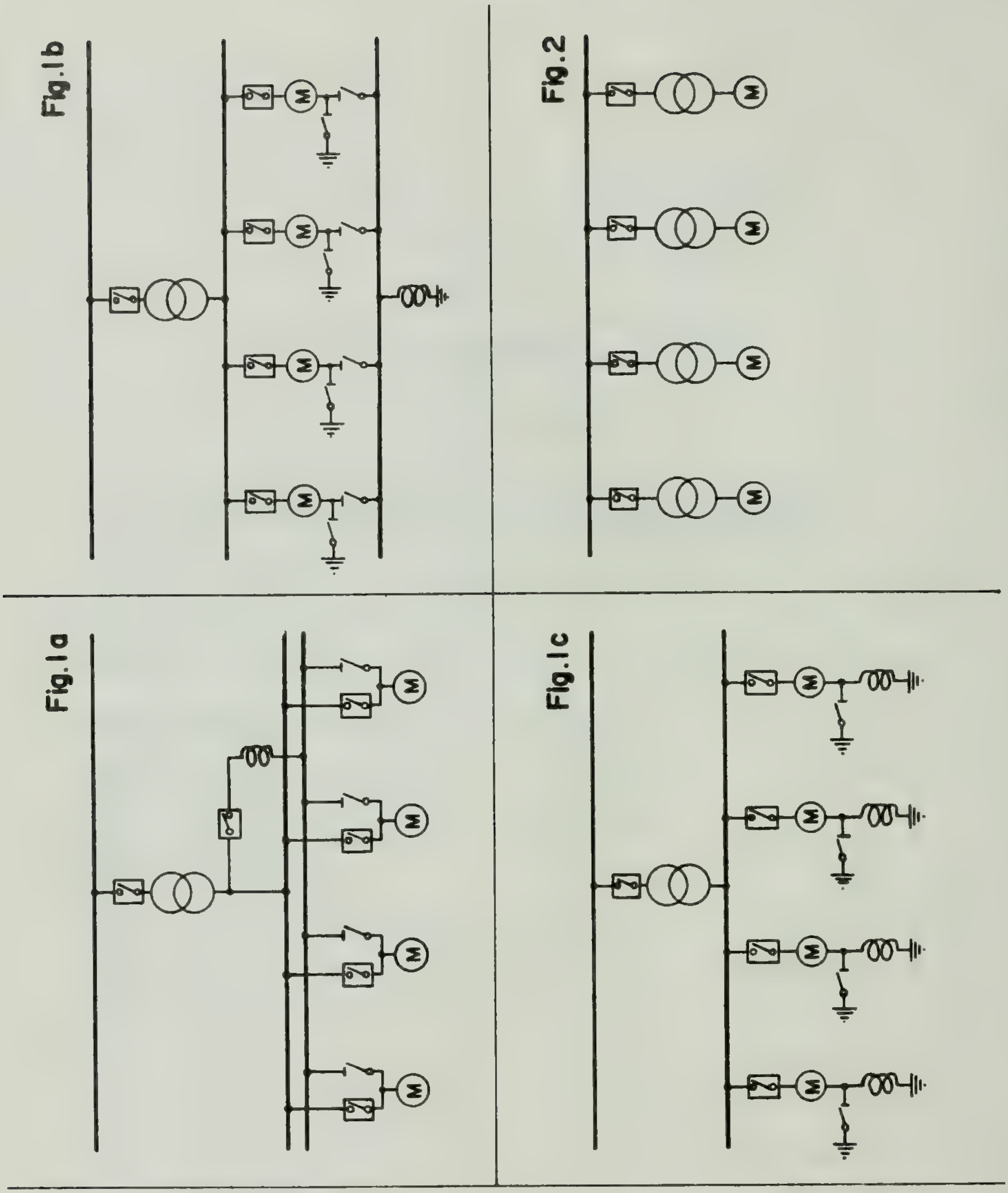
Exhibit 8 : Comparison of Starting Methods

Tehachapi Motor Study   Motor-Columbus Ltd.

Large high-speed Synchronous Machines
of
Swiss Manufacture

The list contains (not necessarily all) synchronous machines of more than 30,000 kVA power and speeds of 500 rpm or more, which had been ordered with Swiss manufacturers since 1945.

Item	Name of power station	Country	No.of units	Shaft pos.	Rated rpm	Rated output in kVA	Year of order
1	Vinodol	Jugoslavia	3	H	500	35,000	1951-53
2	Salanfe	Switzerl.	2	H	500	37,500	1948
3	Lyse	Norway	3	H	500	38,000	1948-52
4	Castasegna	Switzerl.	4	H	500	38,000	1955
5	Maar	Norway	3	H	500	40,000	1946-51
6	Granfeste	Norway	2	H	500	45,000	1955-56
7	Fortun	Norway	2	H	500	45,000	1958
8	Monte Argento	Italy	3	V	500	46,000	1951
9	Vinstra	Norway	4	V	500	50,000	1947-55
10	Lyse	Norway	3	H	500	50,000	1955-61
11	Göschenen	Switzerl.	2	V	500	50,000	1956
12	Hol I	Norway	2	V	500	55,000	1953
13	NEA	Norway	3	V	500	55,000	1956
14	Ixtapantongo	Mexico	1	V	500	57,500	1950
15*	Bärenburg	Switzerl.	4	V	500	64,000	1957
16	Riddes	Switzerl.	5	H	500	67,000	1951



Tehachapi — Motor Study **exhibit 3**

Possible System Arrangements

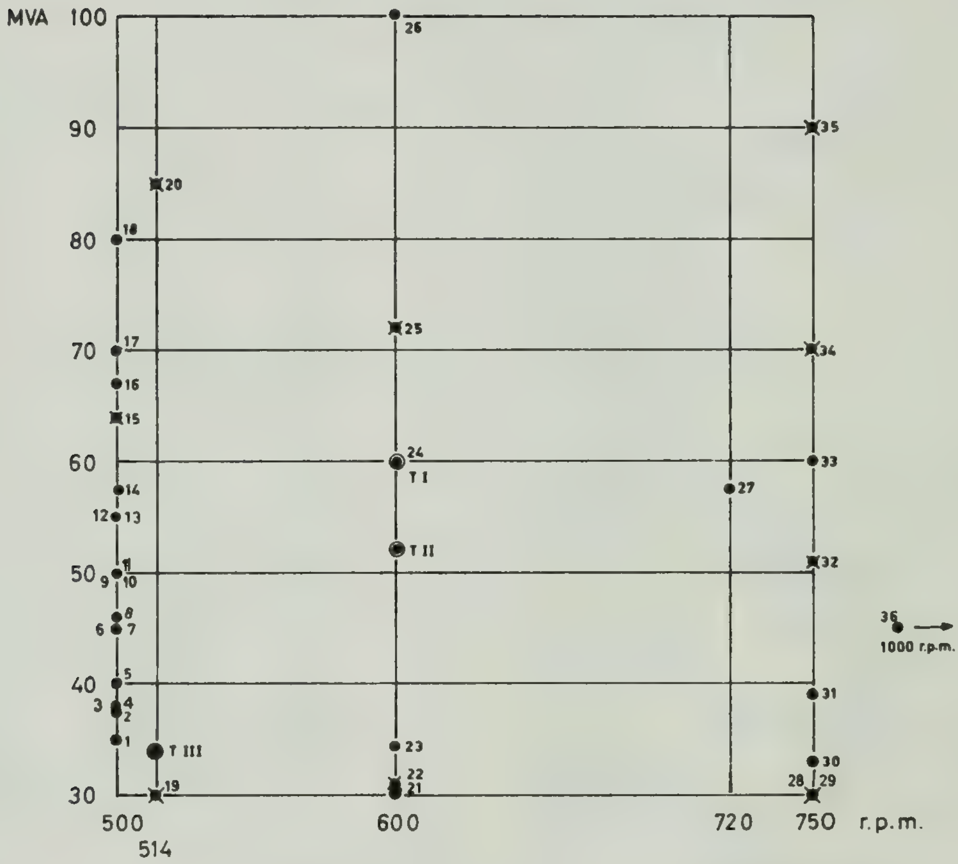
MOTOR-COLUMBUS AKT.-GES. FÜR ELEKTRISCHE UNTERNEHMUNGEN BADEN SCHWEIZ	MASSTAB	F	E	D	DATUM 15.2.65	tu NI
	/	C	B	A	No. 3.24.25.05.1	
		Änderungen				

Item	Name of power station	Country	No.of units	Shaft pos.	Rated rpm	Rated output in kVA	Year of order
17	Avise	Italy	1	H	500	70,000	1953
18	Nendaz	Switzerl.	6	H	500	80,000	1956-63
19*	Moyopampa	Peru	3	H	514	30,000	1947-53
20*	Huinco	Peru	4	H	514	85,000	1960-62
21	El Cobano	Mexico	2	V	600	30,000	1951
22*	Verbano	Switzerl.	4	V	600	32,000	1950
23	Croix	Switzerl.	2	H	600	34,500	1956
24	Paradela	Portugal	1	V	600	60,000	1953
25*	Sils	Switzerl.	4	V	600	72,000	1957
26	Tierfehd	Switzerl.	3	H	600	100,000	1958
27	Tingambato	Mexico	3	V	720	57,500	1952
28	Wassen	Switzerl.	2	V	750	30,000	1946
29*	Piottino	Switzerl.	1	V	750	30,000	1955
30	L'Hospitalet	France	3	H	750	33,000	1955
31	Hemsil I	Norway	1	V	750	39,000	1956
32*	Safien-Platz	Switzerl.	2	V	750	51,000	1954
33	Fionnay	Switzerl.	3	V	750	60,000	1951-55
34*	Ferrera	Switzerl.	3	H	750	70,000	1958
35*	Pradella	Switzerl.	4	V	750	90,000	1965
36	Robiei	Switzerl.	4	V	1000	45,000	1963

H = horizontal shaft
V = vertical shaft

*) Consulting Engineer Motor-Columbus

February 16, 1965
MC 2-Lp/bch



• Large high-speed salient pole synchronous machines designed and built by Swiss manufacturers

$P \geq 30$ MVA, $n \geq 500$ r.p.m.

(The numbers correspond to the items of the respective reference list)
(✖ built under the supervision of Motor-Columbus)

◉ Tehachapi alternatives

TEHACHAPI – MOTOR STUDY

MOTOR-COLUMBUS AKT.-GES. FÜR ELEKTRISCHE UNTERNEHMUNGEN BADEN SCHWEIZ	MASSTAB /	F E D C B A Änderungen	DATUM 10.2.65 No. 224.25.02

EXHIBIT 4

Maschinenfabrik Oerlikon Zürich

Postanschrift: Maschinenfabrik Oerlikon, Postfach 8050 Zürich

Telephon: (051) 48 18 10
Telex: 52135 Oerlik Zurich
Telegramme: Oerlik Zurich
Postcheck: Zürich 80 - 671

MOTOR - COLUMBUS A.G.
für Elektrische Unternehmungen

5401 Baden

14 JAN 1965 09610

Ihr Zeichen	Ihr Schreiben vom	Unser Zeichen	Zürich, Affolternstrasse 52
S-Hm/btt	8.12.64	VM/Ro/ck	13. Januar 1965

Projekt Tehachapi USA
Synchronmotoren für Pumpenantrieb

Wir beziehen uns auf Ihren Brief vom 8. Dezember 1964 und auf die Besprechung bei uns am 17. Dezember, an der Ihrerseits die Herren Hartmann, Leupin und Weber teilnahmen. Der getroffenen Abmachung entsprechend senden wir Ihnen beiliegend ein Exposé, welches die von Ihnen gestellten Fragen behandelt.

Sie erklärten sich bereit, die Uebersetzung dieses Berichtes ins Englische selber auszuführen. Wir wären Ihnen sehr dankbar, wenn Sie uns eine Kopie der Uebersetzung zur Verfügung stellen könnten.

Im übrigen hoffen wir, dass der Inhalt unseres Exposés Ihren Erwartungen entspricht. Wir würden uns freuen, wenn das Projekt in der erwarteten Art sich entwickeln würde und grüssen Sie

hochachtungsvoll
MASCHINENFABRIK OERLIKON

2

Beilage:
Exposé, 2-fach

309

BM 1371

KM/S/sch
11.1.1965

Betr. Projekt Fehachapi U.S.A
Synchronmotoren für Pumpenantrieb

1. Allgemeines zur Konstruktion

Bei den vorgesehenen Synchronmotoren von 80'000 bzw. 70'000 HP bei 600 U./min 60 Hz handelt es sich um Maschinen, die bei Nenndrehzahl bereits die hohe Umfangsgeschwindigkeit von ca. 90 m/sec aufweisen. Im Vergleich zu Wasserkraftgeneratoren ähnlicher Grösse erreichen die mechanischen Beanspruchungen trotzdem noch keine extremen Werte, weil die Durchbrenndrehzahl nur 40 % über der Nenndrehzahl liegt; das ergibt im Durchgang eine Geschwindigkeit von ca. 125 m/sec. Mit den heute üblichen Konstruktionen und den seit einigen Jahren erhältlichen Stahlqualitäten treten erst bei Umfangsgeschwindigkeiten im Durchgang von mehr als 160 m/sec schwierige Probleme für die Polbefestigung und die Gestaltung des Rotorkörpers auf. Es kann dann z.B. nötig werden, wegen der auftretenden mechanischen Beanspruchungen die Polkörper massiv, in Stahlguss oder geschmiedet, auszuführen. Im vorliegenden Fall trifft das jedoch nicht zu; im Hinblick auf die Festigkeit ist der Konstrukteur frei, die Pole entweder massiv oder lamelliert auszuführen. Dies gilt natürlich erst recht für die Motoren gemäss Variante 3, die 46'000 HP bei 514 U./min leisten sollen.

Der allgemeine Aufbau der Motoren würde sich von den zahlreichen in Betrieb befindlichen vertikalen Hydrogeneratoren kaum unterscheiden. Die Maschinen können über Boden oder versenkt angeordnet werden; dies hängt von der baulichen Gesamtdisposition, insbesondere für den hydraulischen Teil der Anlage, ab.

Die Kühlerelemente werden am Umfang des Statorgehäuses angebracht. Bei der ziemlich hohen Kühlwassertemperatur von 87 ^{o}F = 30,5 ^{o}C empfiehlt es sich, eine Kaltlufttemperatur von 45^{o} zuzulassen, damit die Kühler nicht unwirtschaftlich gross werden. Die Kupfertemperatur würde dann im Maximum 45 + 60 = 105 ^{o}C erreichen; dieser Wert liegt immer noch 15 ^{o}C tiefer als die 120 ^{o}C, die von jeder nach Klasse B isolierten Wicklung ohne Beeinträchtigung der Lebensdauer ausgehalten werden.

Es ist für Synchronmotoren vorteilhaft, wie in der provisorischen Spezifikation vorgesehen, den cos φ auf 1,0 festzusetzen. Dies ergibt die Maschine mit der kleinsten Scheinleistung, also auch dem niedrigsten Preis, und dem besten Wirkungsgrad. Letzterer beläuft sich auf ca. 98,5 %.

Der 80'000 HP-Motor lässt sich für die vorgeschriebene Erwärmung von 60 ^{o}C nach unserer Erfahrung mit einem GD^2 von ca. 230 t m^2 ausführen; für die beiden anderen

Varianten liegt der Wert noch etwas tiefer. Ein kleineres Schwungmoment erleichtert natürlich den Hochlauf, da die infolge der Beschleunigung auftretende Wärmemenge im gleichen Verhältnis zurückgeht.

2. Anlaufwärme und Polkonstruktion

Der Anlauf stellt das einzige wirklich schwierige Problem bei Synchronmotoren dieser Grösse dar. Dies gilt sogar dann, wenn die Pumpe während des Anlaufs entleert wird und somit nur ein vernachlässigbar kleines Gegenmoment entwickelt. Denn die Schwierigkeit besteht nicht darin, mit dem Motor das nötige Anlaufmoment zu erreichen, sondern die beim Hochlauf entstehende Wärme im Rotor aufzunehmen, ohne dass irgend ein Teil desselben Schaden nimmt.

Die Wärmemenge Q, die nur durch die Beschleunigung der Schwungmasse, ohne jedes Gegenmoment, während eines vollständigen Hochlaufs im Rotor entsteht, ist gleich gross wie die lebendige Energie des Rotors bei voller Drehzahl. Somit gilt

$$Q = \frac{GD^2}{8\,g} \cdot \left(\frac{\pi \cdot n}{30}\right)^2 \cdot \frac{1}{427} \text{ Kcal.}$$

Für den 80'000 HP-Motor (Totales $GD^2 = 230 + 32 = 262$ tm^2) ergibt sich

$$Q = \frac{262 \cdot 10^3}{8 \cdot 9{,}81} \cdot \left(\frac{\pi \cdot 600}{30}\right)^2 \cdot \frac{1}{427} = 31000 \text{ Kcal.}$$

Würde man den Motor mit lamellierten Polen und Dämpferwicklung ausrüsten, wie es bei Motoren geringerer Leistung üblich ist, so müssten schätzungsweise 80 % obiger Wärmemenge, d.h. 24800 Kcal. von der Dämpferwicklung aufgenommen werden; der Rest verteilt sich auf Polspulen und Anlasswiderstand. Eine genaue Untersuchung zeigt, dass bei den durch die gesamten Abmessungen der Maschine gegebenen Platzverhältnissen in einem Polschuh höchstens 4000 mm^2 Dämpferstab-Querschnitt untergebracht werden könnte. Die Stablänge beträgt etwa 1,80 m. Nimmt man als Material Bronce an ($\gamma = 8{,}8$) und rechnet mit 25 % Zuschlag für die stirnseitigen Verbindungen, so ergibt sich ein Gewicht der Dämpferwicklung von

$$1{,}25 \cdot 12 \cdot 4000 \cdot 1{,}8 \cdot 8{,}8 \cdot 10^{-3} = 950 \text{ kg.}$$

Hieraus folgt bei einer spezifischen Wärme von 0,09 Kcal/kg, °C eine Uebertemperatur der Anlaufwicklung von

$$\frac{24800}{950 \cdot 0{,}09} = 290\ ^\circ\mathrm{C}$$

Bei direktem Einschalten würde die Hochlaufzeit bis zur annähernd vollen Drehzahl ohne Gegenmoment der Pumpe nur etwa 6-10 s betragen. In dieser kurzen Zeit erfolgt keine ins Gewicht fallende Wärmeabgabe an das Eisen des Polkörpers. Anderseits

erhöht sich die Anlaufzeit und damit die aufgespeicnerte Wärmemenge, sobald der Motor nicht nur die Schwungmasse zu beschleunigen, sondern ausserdem ein Gegenmoment der Pumpe zu überwinden hat. Dies trifft im vorliegenden Fall zu. Die gesamte Erwärmung wird sich daher auf mindestens 400 °C belaufen. Das hat eine Wärmedehnung in der Grössenordnung von 15 mm zur Folge, wobei die Erwärmung und damit die Ausdehnung der einzelnen Stäbe in der Regel nicht unbeträchtliche Unterschiede aufweist. Zudem wird die Festigkeit auch guter Broncen bei so hohen Temperaturen bereits merklich geschwächt. Es ist leicht einzusehen, dass es praktisch unmöglich ist, für solche Verhältnisse eine betriebssichere ~~zuverlässige~~ Dämpferwicklung zu konstruieren. Für die Motoren der Variante 3 liegen die Verhältnisse ein wenig günstiger; doch sind auch hier Temperaturen zwischen 350 und 400 °C mit Sicherheit zu erwarten. Es kann daher für keine der für das vorliegende Projekt in Betracht kommenden Varianten verantwortet werden, die Motoren mit geblechten Polen und Dämpferwicklung auszuführen.

Bei massiven Polen dagegen verteilt sich die Wärme, obwohl sie nur in einer relativ dünnen Oberflächenschicht des Polschuhs entsteht, sehr rasch über eine grössere Zone. Schätzt man die Dicke derselben sehr vorsichtig nur auf 20 mm, so kommt man auf eine Erwärmung von ca. 130°. Diese wirft in dem massiven Polkörper keinerlei Probleme auf. Die Ausführung mit massiven Polen aus Stahlguss oder geschmiedetem Stahl, stellt somit für Synchronmotoren dieser Grösse mit Selbstanlauf die einzige betriebssichere Lösung dar.

Die rechnerische Behandlung des Anlaufvorganges bei Motoren mit Massivpolen ist schwieriger und die Ergebnisse sind weniger zuverlässig, als dies bei Motoren mit Anlaufwicklung der Fall ist. Es ist daher wichtig, dass die europäische, vor allem auch die schweizerische Industrie zur Kontrolle der Rechnung zahlreiche Messungen an ausgeführten Maschinen durchgeführt hat und somit über ein umfangreiches Erfahrungsmaterial verfügt. Dieser Umstand gestattet den massgebenden Firmen, Synchronmotoren dieser Grösse auch mit massiven Polen so zu entwerfen, dass auch hinsichtlich der Anlaufeigenschaften ein Optimum erreicht wird.

3. Schaltung

Durch die Art der Disposition des elektrischen Teils der ganzen Anlage lässt sich der Anlaufvorgang weitgehend beeinflussen. Gemäss vorläufiger Spezifikation ist vorgesehen, je 4 bzw. 3 Motoren auf einen gemeinsamen Transformator zu schalten und die Motoren direkt anlaufen zu lassen. Mit der Eigenreaktanz des Motors von maximal 19 % und der anteiligen Reaktanz des gemeinsamen Transformators von ca. 3 % ergibt sich eine Gesamtreaktanz von ca. 22 % und ein Anlaufstrom von $\frac{100}{22} = 4,5\ I_n$. Dies

entspricht für den 80'000 HP-Motor einer Anlaufscheinleistung im Netz von 270 MVA und bei 4000 MVA Kurzschlussleistung des Netzes einem Spannungsabfall von 6,75 %. Die gewünschte Grenze von 5 % lässt sich also mit dieser Schaltung nicht einhalten. Zum mindestens müsste eine Drosselspule vorgesehen werden, die gegen Ende des Hochlaufs kurzgeschlossen wird. Sie wäre so zu bemessen, dass sie den Einschaltstrom auf das 3,33-fache und damit die Einschaltleistung auf 200 MVA begrenzt. Bei Anordnung der nötigen Trenner würde für jede Gruppe von 4 bzw. 3 Motoren eine Drosselspul genügen. Bei der 46'000 HP-Variante kann sie wegfallen.

Neben der vorgeschlagenen Schaltung sollten auch noch andere Lösungen grundlich studiert werden. Wenn z.B. jeder Motor mit einem eigenen Transformator in Blockschaltung verbunden würde, so wären im Falle der Variante 1 (80'000 HP) 17 Trafos à 62 MVA anstatt 5 à 250 MVA erforderlich; dabei wurde in jedem Fall 1 komplette Einheit als Reserve angenommen. Für die Trafos allein wäre diese Lösung zweifellos teurer. Für die Gesamtanlage sieht es aber vielleicht anders aus; denn das ganze Sammelschienen-System nebst Trennmessern, das bei Gruppenschaltung nötig ist, kann bei Blockschaltung fortfallen. Auch eine Drosselspule wird dann überflüssig, weil die Gesamtreaktanz (Motor 19 % + Trafo 11 %) genügt, um die Anlaufleistung auf 200 MVA zu begrenzen. Als weiterer Vorteil dieser Lösung darf gelten, dass bei einem Defekt an einem Trafo nur ein einziger Motor ausfällt und nicht deren drei oder vier.

4. Weitere Anlaufprobleme

Für den Motor selbst wäre der Anlauf an voller Spannung ohne weiteres zulässig. Die Wickelköpfe der Statorwicklung könnten genügend stark abgestützt werden, um selbst den 4,5-fachen Strom einmal pro Tag auszuhalten, ohne sich zu deformieren. Ob die Lebensdauer der Statorwicklung durch die auftretenden mechanischen Beanspruchungen überhaupt merklich herabgesetzt wird, lässt sich schwer abschätzen; zahlenmässige Angaben darüber sind jedenfalls nicht möglich. Wohl aber ist damit zu rechnen, dass die Bandagen schneller als sonst locker werden. Demgemäss müssten die periodischen Revisionen in um so kürzeren Zeitintervallen vorgenommen werden, je grösser die beim Einschalten auftretenden Stromstösse sind; dies hängt wiederum von der gewählten Schaltung ab.

Thermisch wird die Statorwicklung durch die Ueberströme beim Hochlauf nicht nennenswert beansprucht; die Temperatur-Erhöhung hat die Grössenordnung von etwa 20 °C.

Das Synchronisieren des Motors gegen ein Pumpenmoment von maximal 62 % ist auch bei massiven Polen ohne Schwierigkeit möglich, wenn im Polradkreis der Maschine ein Anlasswiderstand passender Grösse vorgesehen wird.

5. Zusammenfassung

Folgende drei Feststellungen bilden die Schlussfolgerungen aus dem Gesagten:

- Die allgemeine Konstruktion der Maschine bietet keine grossen Schwierigkeiten.
- Massive Pole sind unerlässlich.
- Blockschaltung Motor-Transformator wird dringend empfohlen.

Seefeld.

T r a n s l a t i o n

(letter of Maschinenfabrik Oerlikon, Zurich) January 21, 1965

Motor-Columbus AG
für elektrische Unternehmungen

5401 B a d e n

Project Tehachapi USA
Synchronous Motors as Pump drive

Referring to your letter of December 8, 1964, and to the discussions held with your Messrs. Hartmann, Leupin, and Weber we are enclosing herewith a write-up dealing with the questions raised by you.

You declared your willingness to translate this write-up into English. We should be very grateful if you would send us a copie of your translation.

We hope that the contents of our write-up corresponds to your expectations and should be glad if the project develops in the direction expected.

We remain,

Very truly yours,
MASCHINENFABRIK OERLIKON

Encl.:
Write-up
(in duplicate)

EXHIBIT 5

Translation

Re: Tehachapi
Synchronous Motors for Pumps

KM/S/sch
11.1.1965

1. General remarks concerning construction

The considered synchronous motors of 80,000 and 70,000 HP respectively, at 600 rpm, 60 cps, are motors which show at rated speed the high circumferential velocity of approx. 90 m/s. Nevertheless, compared with hydro-power generators of similar size, the mechanical stresses do not reach extreme values, because the run-away speed only amounts to 140 % of the rated speed, resulting in a circumferential velocity of approx. 125 m/s. With the constructions used nowadays and the steel qualities which are available since some years, serious problems concerning the pole fixation and the design of the rotor body arise only at circumferential velocities of more than 160 m/s. Because of the deriving mechanical stresses it may then become necessary to manufacture the pole bodies in solid forged or cast steel. This, however, is not the case here; with regard to the stresses, the designing engineer is free to make the poles either solid or laminated. Of course, this is all the more valid for the motors according to alternative 3, which have an output of 46,000 HP at 514 rpm.

The general structure of the motors would not much differ from the numerous vertical hydro-generators in operation. The machines can be placed above the floor or sunk; this depends on the structural general arrangement, especially concerning the hydraulical part of the station.

The air coolers are arranged on the circumference of the stator frame. It is recommended to allow a cooling air temperature of 45 °C because of the rather high cooling water temperature of 87 °F = 30.5 °C, so that the coolers do not get uneconomically big. The copper temperature would then reach maximum 45 + 60 = 105 °C; this value is still 15 °C lower than the temperature of 120 °C, which is withstood by any winding isolated according to class B, without shortening the lifetime.

As provided in the preliminary specification, it would be favourable to stipulate 1.0 power factor. This results in a machine with the smallest apparent power, and, consequently, lowest price and best efficiency. The latter amounts to approx. 98.5 %.

According to our experience, the 80,000 HP-motor can be laid out for the required temperature rise of 60 °C with a GD^2 of 230 t m2*; for the two other alternatives the value is somewhat lower. Of course, a smaller GD^2 facilitates the starting as the quantity of heat, due to acceleration, retrogrades proportionally.

2. Starting heat and pole design

Starting is the only really difficult problem with synchronous motors of this size. This is true even with the pump being dewatered during starting and, consequently, developing only a negligible small reactive torque, because the problem is not how to reach the necessary starting torque, but how to absorb the heat developing in the rotor during starting without damaging any part thereof.

---

*) $\dfrac{GD^2\ (tm^2)}{WR^2\ (lb.ft^2)} = \dfrac{1}{6}$

The quantity of heat Q generated in the rotor by accelerating the flywheel mass only, without any outside reactive torque, is as big as the kinetic energy of the rotor at full speed. Therefore:-

$$Q = \frac{GD^2}{8\ g} \times \left(\frac{\pi n}{30} \right)^2 \times \frac{1}{427} \quad \text{Kcal}$$

For the 80,000 HP-motor (total GD^2 = 230 + 32 = 262 tm^2):-

$$Q = \frac{262 \times 10^3}{8 \times 9.81} \times \left(\frac{\pi \times 600}{30} \right)^2 \times \frac{1}{427} = 31{,}000 \quad \text{Kcal}$$

If the motor is equipped with laminated poles and amortisseur winding, as usually done for motors with lower output, about 80 % of the above mentioned heat, i.e. 24,800 Kcal would have to be absorbed by the amortisseur winding; the rest is divided on the pole coils and starting resistor. A thorough investigation shows that according to available space only 4000 mm2 amortisseur rod sectional area can be placed in one pole shoe. The length of the rod is about 1.80 m. If bronze is chosen (= 8.8) and if an additional weight of 25 % for the frontal connections is assumed, the weight of the amortisseur winding will be

$$1.25 \times 12 \times 4000 \times 1.8 \times 8.8 \times 10^{-3} = 950 \text{ kg}$$

Hence, with a specific heat of 0.09 Kcal/kg °C an overtemperature of the amortisseur winding results in

$$\frac{24800}{950 \times 0.09} = 290\ ^{\circ}C$$

When starting under full voltage, the starting time up to nearly full speed would be only about 6 - 10 s, without an reactive torque of the pump. During this short period, no serious heat transfer to the steel of the pole body can take place. On the other hand the starting time as well as the generated heat quantity

are increased if the motor has not only to accelerate its flywheel mass, but also to overcome a reactive torque of the pump. This is the case here. The total temperature rise will, therefore, reach at least 400 °C. Subsequently, there will be a thermal expansion of 15 mm, whereby the heating, and consequently the expansion of the individual rods show considerable differences. Furthermore, the strength of bronze even of good quality will be considerably reduced at such high temperatures. It is obvious that it is practically impossible to design a reliable amortisseur winding for such operating conditions. Concerning the motors of alternative 3, the operating conditions are somewhat more favourable; but also in this case temperatures between 350 and 400 °C must be expected. Therefore, one cannot take the responsibility to equip the motors with laminated poles and amortisseur windings for either alternative taken into consideration for the actual project.

With solid poles, however, the heat spreads very fast across a larger area, although it is developed only in a relatively thin surface layer of the pole shoe. If the thickness of same is estimated to be only 20 mm, there will be a temperature rise of approx. 130 °C. This does not raise any problems in the solid pole body. Therefore, the only reliable solution for self-starting synchronous motors of this size is the design with solid poles of cast steel or forged steel.

The calculation of the starting process of motors with solid poles is more difficult and the results are less reliable than with motors with amortisseur winding. Therefore, it is very important that the European, especially the Swiss industry has made numerous measurement on already existing units in order to control the calculation, thus disposing of a broad experience. This enables the leading manufacturers to design synchronous motors of this size also with solid poles in such a way, that an optimum in reached also concerning the starting characteristics.

3. Switching

Starting can be considerably influenced by the layout of the electrical part of the whole station. According to the preliminary specification, it is provided to connect 3 or 4 motors to one common transformer and to start the motors under full voltage. Self reactance of the motor of max. 19 % and corresponding reactance of the common transformer of approx. 3 % result in a total reactance of approx. 22 % and an inrush current of $\frac{100}{22} = 4.5\ I_n$. Concerning the 80,000 HP-motor this corresponds to a starting apparent power of 270 MVA and with 4000 MVA short-circuit power of the system to a voltage drop of 6.75 %. This arrangement does not allow to restrict the voltage drop to the required limit of 5 %. At least an impedance coil should be provided which will be short-circuited at the end of starting. This impedance coil would have to be laid out in such a way, that it limits the inrush current to the 3.33 I_n and, consequently, the starting power to 200 MVA. When providing the necessary breakers, one impedance coil would be sufficient for each group of 4 or 3 motors. At the alternative of 46,000 HP it can be omitted.

Apart from the suggested arrangement, also other solutions should be studied thoroughly. If, e.g. every motor would be connected to its own transformer (block arrangement) it would be necessary to have 17 transformers à 62 MVA instead of 5 à 250 MVA for alternative 1 (80,000 HP); hereby always taken into consideration one complete transformer is meant to be a spare. No doubt, this solution would become more expensive for the transformers alone.
It may, however, be different if one takes the whole conception into consideration, because the complete busbar-system including disconnecting switches necessary for group arrangement can be omitted when using block arrangement. Even an impedance coil becomes superfluous, because in order to limit the starting power to 200 MVA a total reactance (motor 19 % + transformer 11 %) is

quite sufficient. Another advantage of this solution is the fact, that in case of a transformer failure only one motor fails and not three or four.

4. Further starting problems

For the motor itself a starting at full voltage would be permittable. The end turns of the stator winding could be braced in a way to withstand even an inrush current of 4.5 I_n once per day without deforming. Whether the lifetime of the stator winding will be considerably reduced by the mechanical stresses is very difficult to say; in any case, it is impossible to give information in figures. However, it must be expected that the bracings become loose in a shorter time than normally experienced. Therefore, with higher inrush current, the periodical overhauls would have to be carried out in shorter time intervals. This again depends on the switching arrangement chosen.

There is no considerable thermal stress on the stator winding by overcurrents during starting; the temperature rise is about 20 °C. Also synchronizing of the motor against a pump torque of maximum 62 % is possible with solid poles without any difficulty, if a resistor of a matching size is provided in the rotor field circuit of the machine.

5. Conclusions

The conclusions of the above mentioned are as follows:-

- The general design of the motors does not raise any major problems.
- Solid poles are indispensable.
- Block arrangement motor - transformer is strongly recommended.

signed: Seefeld

February 11, 1965
MC - S-Hm/2-Lp/bch

SOCIÉTÉ ANONYME DES EXHIBIT 6

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GENÈVE, le ~~13 janvier~~ 1965

14 JAN 1965 00624

MOTOR-COLUMBUS
AG für elektrische
Unternehmungen

5401 Baden

OBJET:

Projet Tehachapi USA
Moteurs synchrones pour
l'entrainement de pompes

Votre réf.: S-Hm/btt Votre lettre du 8.12.1964 Notre référence DP/Gi/dh
A rappeler dans votre réponse s. v. p.

Messieurs,

Nous avons bien reçu votre lettre du 8 décembre 1964 qui a retenu notre meilleure attention. Malheureusement, à cause de la longue fermeture de notre usine pour les fêtes de fin d'année, il ne nous a pas été possible de vous répondre plus rapidement.

Tout en vous priant de bien vouloir nous en excuser, nous avons maintenant l'avantage de pouvoir répondre, dans le même ordre que vous avez adopté, aux diverses questions contenues dans votre lettre.

1. La construction de moteurs de 80'000 ch à 600 t/mn et 40 % de survitesse ne présente aucune difficulté particulière. La roue polaire consiste en un moyeu central en acier coulé sur lequel viennent se fixer 2 demi-arbres avec plateaux de fixation. La jante rotorique est constituée par des anneaux en acier forgé frettés sur le moyeu. Les pôles peuvent être du type feuilleté avec amortisseurs pour démarrage adossé à fréquence variable, ou massifs avec connexions entre pôles pour démarrage en asynchrone à tension réduite ou démarrage à fréquence variable.

2. Jusqu'à présent, les gros moteurs synchrones bâtis par notre société sont prévus pour démarrage à roue noyée: couple de démarrage C_d à l'enclenchement env. 15 %. C_{ds} avant synchronisation 55 % (60 % serait également possible si nécessaire).

3. Nous recommandons l'usage des pôles massifs avec connexions entre pôles comme la solution la plus robuste pour ce genre de service.

Annexé:

./.

Form. 47 c 6. 63 30000

4.-5. Pour de telles puissances, le démarrage direct sous pleine tension est exclu du fait des répercussions sur le réseau (I_d env. 5 à 6 x I_n, cosphi$_d$ env. 0,2) et des contraintes électrodynamiques répétées sur les développantes.

6. Démarrage à tension réduite par autotransformateur à gradins et insérateur de prises ou démarrage dos à dos à fréquence variable avec groupe de lancement unique pour toute la centrale (puissance env. 1/10 à 1/6 de la puissance apparente des groupes principaux) ou utilisation de l'un des groupes principaux si les groupes sont réversibles, la puissance disponible en pompage étant alors inférieure d'un groupe à la puissance en turbinage.

Nous espérons ainsi vous avoir rendu service et restons a votre disposition pour tout renseignement complémentaire qui pourrait encore vous être utile.

Nous vous prions d'agréer, Messieurs, nos salutations distinguées.

S.A. DES ATELIERS DE SECHERON

ppa. Giaminatti

Form. 49 5.64 10 000

EXHIBIT 7

Ateliers de Sécheron SA
G e n è v e

letter of January 13, 1965

T r a n s l a t i o n

MOTOR-COLUMBUS
AG für elektrische Unternehmungen

5401 B a d e n

Project Tehachapi USA
Synchronous Motors as Pump drive

Gentlemen:

Thank you for your letter of December 13, 1965. Our answer has been delayed because our works were closed during the end of year's holidays. Please excuse us.

The answers to your questions are presented in the same order as that used in your letter.

1. The construction of 80,000 HP at 600 rpm and 40 % overspeed motors presents no particular difficulties. The pole wheel consists of a cast steel core to which two half shafts with fixation end plates are attached. The rotor's outside rim is made of forged steel rings shrunk on the core. The poles can be laminated with amortisseur windings for back to back synchronous starting, or solid with interconnections from

pole to pole for asynchronous starting with reduced voltage or back to back synchronous starting.

2. Up to now, the large synchronous motors built by our company are foreseen for starting with the pumps filled with water: break-away torque C_d approx. 15 %, pull-in torque C_{ds} 55 % (60 % would be possible, if necessary).

3. We recommend the use of solid poles with pole to pole interconnection as the most rigid solution for this type of application.

4/5. For this range of power, full voltage starting is impossible because of reactions on the grid (inrush current I_d is approx. equal 5 to 6 times the rated current I_n, starting power factor approx. 0.2) and because of repeated electrodynamic stresses on the end turns of the stator winding.

6. (Possible starting methods are:-)*
Reduced voltage starting with stepping autotransformer, or back to back synchronous starting with a single starting group for the whole plant (power approx. 1/10 to 1/6 of the apparent power of the main units) or the use of one of the main units, if these are reversible, however, the available pumping capacity is then reduced by one unit compared with the possible turbine capacity.

We hope that our answer will be of good use to you,and we remain at your disposal for any additional questions.

Very truly yours,

SA DES ATELIERS DE SECHERON

* Insert of translator

COMPARISON OF STARTING METHODS

	A. Pump filled with water	Pump dewatered
1. Pump	Advantages: - usual method - hydraulically simple Disadvantages: - high reactive pump torque (50 - 60 % rated torque at pull-in)	Advantages: - low reactive pump torque (2 - 5 %) at pull-in - well established method with two-stage, double-flow and single-stage pumps Disadvantages: - novel method for four-stage pumps, feasibility to be investigated further - hydraulically complicated, more complicated controls - lower pump efficiency due to larger wear ring clearances
2. Main motor starting		
2.1 full voltage	Advantages: - electrically simple and cheap Disadvantages: - problems with stator winding due to high inrush current - problems with heating of amortisseur winding, if such is used - voltage drop on network	Advantages: - electrically simple and cheap Disadvantages: - problems with stator winding due to high inrush current - problems with heating of amortisseur winding, if such is used, are reduced, but still serious - voltage drop on network
2.2 reduced voltage	Advantages: - reduced inrush current, advantageous for stator winding and network voltage drop Disadvantages: - more complicated and expensive switchgear - problems with heating of amortisseur winding, if such is used	Advantages: - reduced inrush current, advantageous for stator winding and network voltage drop Disadvantages: - more complicated and expensive switchgear - problems with heating of amortisseur winding, if such is used, are reduced, but still serious
3. Starting pony motor on main unit shaft	Technically and economically unfeasible because of pony motor power 50 - 60 % of main motor rating	Advantages: - well established method - pony motor approx. 10 % of main unit power - main motor starting problems eliminated Disadvantages: - additional cost and complication - additional length of unit may require additional bearing - permanent windage losses of pony motor - [illegible]

(continued)

	A. Pump filled with water	Pump dewatered
Starting Pelton turbine on main unit shaft	Technically and economically unfeasible because of starting turbine power 50 - 60 % of main motor rating	Advantages: - main motor starting problems eliminated Disadvantages: - additional cost and complication - additional length of unit - possibly additional bearing - permanent windage loss of starting turbine
5. Torque converter between pump and motor	Advantages: - starting conditions as listed under A 'pump filled with water' are transformed to those listed under B 'pump dewatered' Disadvantages: - additional cost and complication - additional length of unit - increased building height - additional pump thrust bearing - permanent windage losses of torque converter	redundant
6. Synchronous starting		
6.1 with main pump-motor unit used as turbine-generator starting unit	Advantages: - eliminates main motor starting problems Disadvantages: - one additional unit required - additional 3-phase starting bus and switching system required - separate exitation system required for starting - complicated and time-consuming operation - turbine output possibly not sufficient to develop 60 % pull-in torque	Advantages and Disadvantages: - same as under 6.1.A, but ample power available
6.2 with special turbine-generator starting unit	Advantages: - eliminates main motor starting problems Disadvantages: - rating of starting unit 50 - 60 % of main unit power - two such starting units required, one as spare - additional 3-phase starting bus and switching system required - separate exitation system required for starting - complicated and time-consuming operation	Advantages and Disadvantages: - same as under 6.2.A, but rating of starting unit reduced to approx. 10 % of main unit
6.3 with special motor-generator starting unit	Advantages: - eliminates main motor starting problems Disadvantages: - technical feasibility to be investigated - rating of starting unit 50 - 60 % of main unit power - two of such starting units required, one as spare - separate exitation system required for starting. - complicated and time-consuming operation	Advantages and Disadvantages: - same as 6.3.A, but rating of starting unit reduced to approx. 10 % of main unit

January 21, 1965
S - Hm/bch

EXHIBIT I

CHAPTER 5

VALVE STUDY

A. PURPOSE AND SCOPE

The purpose of the valve study is to develop a better understanding of valve problems and possible solutions to these problems as they relate to Tehachapi design parameters. The study will result in outlines of common valve types and their operating characteristics. It will present analyses of various design alternates with their relative merits. Field experience with operation and maintenance of valves will be described and cost-weight relationships tabulated. All of this information will be used to develop specific valve requirements for Tehachapi pumping applications and to fairly delineate the advantages and disadvantages of valve types which fit these requirements.

The scope of the valve study will include investigations in all fields directly associated with the results mentioned above. Valve operations and applications, valve design and maintenance and valve costs and weights will comprise a major portion of the scope of work. Valve manufacturers will be interviewed and brochures and drawings procured to form a study base. With regard to design and operation, the problems of starting and transient conditions will be researched in the literature and on field trips to hydraulic plants of interest. Field trips and manufacturing plants will provide operation and maintenance data from which valve and operator reliability may be estimated.

B. CRITERIA

The criteria on which the valve study will be based are those which must be met in the functions of the three Tehachapi lift concepts. Operating stresses are probably the most important consideration and these are directly related to operating head and pump discharge diameter. Turbining and water hammer conditions in each lift concept will dictate timing of the valve opening and closing functions and place restraints on operator mechanisms. A virtually continuous mode of operation at Tehachapi will influence valve selection in that simplicity of design and ease of access for maintenance must be considered for overall pumping reliability. Finally, a constant effort must be made to balance technical effectiveness with budget considerations both in initial costs of valves and support structures and in annual expenditures for repair and replacement.

C. VALVE FUNCTIONS AND TYPES

The valve functions are mainly those of isolation for purposes of stopping water backflow or de-watering for maintenance work. The pump inlet valve is used to isolate the pump from the intake pool or from the inlet pipe which feeds it and adjacent pumps at intermediate lift points. Discharge valves may isolate the pump from water in the penstock or may isolate one penstock from the water flowing in another. Any two valves in line may be used to isolate a part of the pumping system for de-watering.

The types of valves applicable to most pumping plant isolation functions are sliding gates, sluice valves, butterfly valves, needle valves, and rotary valves. The construction and operation of these valves will be discussed in the paragraphs below, together with brief mentions of their associated equipment.

The sliding gate is quite simply what its name implies. It is a square or rectangular, large metal plate which normally is used to separate water in the pump draft tube from that in the pumping pool. It is inserted in a guide slot built into the wall of the forebay adjacent to the pumping plant. It may also be installed in a concrete blockhouse at the top of a lift to isolate water in a penstock from that in the surge or storage tanks. It is raised or lowered in its guide slot by chain-and-sprocket hoists which are powered electrically or hydraulically. It may also be lifted by crane hoist. Rollers and counterweights generally facilitate movement. Stop logs are a laminated type of bulkhead similar to a slide gate which may be used as a temporary or emergency closure.

Sluice valves are much like slide gates except that they form an integral part of the piping and are, therefore, usually smaller than slide gates because the valve disc must be housed inside the valve casing. The valve disc is made to slide up and down by the valve shaft and a stuffing box is used to obtain a water seal. The valves may be operated by oil or water pressure or by motor.

Butterfly valves are used extensively in pumping plant installations where they are subjected to low and medium heads. They consist essentially of circular discs which are rotated inside of hollow cylindrical housings. The closed position places the disc face at right angles to the water flow and the open position is at 90° from this angle. The disc is supported on the valve body by means of a valve shaft which penetrates the center of the disc. When fully closed, the valve disc will come into a tight contact with the valve body utilizing a rubber or metal seat ring for water-tightness along the periphery of the disc.

The needle valve is constructed with a spindle shaped interior hull inside a circular streamlined valve body and the flow is stopped or adjusted by moving the conical part of the valve body upstream and downstream. Hydraulic forces acting on the needle are approximately balanced so that the force required to move the needle through its entire travel can readily be provided by mechanical operation. However, large seating forces required by this type of operation makes it more economical to move the needle by internal hydraulic chambers utilizing available head.

Rotary valves are widely used in medium and high head pumping plants as a discharge valve. The rotary valves are of three types: cylindrical, cone or spherical depending on the shape of the plug which is the rotating part. Operation of the cylindrical type is probably easiest to describe since the plug appears much like a segment of the tubing, being of the same inside diameter. This segment of the piping then may simply be turned at right angles to the water flow and valve seats may be made to contact the plug to seal it or withdrawn to allow movement. Plug seating surfaces at both ends of the cylinder are placed equidistant from the center of rotation so that the same seats are used in open or closed positions. Seats may be employed at only one side of the valve or on both upstream and downstream sides with one set being used in emergencies or to facilitate repair of the normal seats. The cone type valve is very similar to the cylindrical type except that it has a tapered shape to its plug walls to facilitate tight seating by a wedging action. The plug in this case moves into contact with the seat rather than mating with movable seats. The spherical valve is built so that a flat plug seating surface is presented to the movable casing seats. Figure 5-I shows details of a typical spherical valve of the type appropriate for use in the pump discharge lines.

D. DESIGN CONSIDERATIONS

Design considerations are mainly those of operating head, valve diameter and ease of installation. The opinions given here on relative merits of all valve types except the sliding gate are opinions offered by manufacturers.

Intake valves for all lift concept pumping pools are subject to low head and must seal draft tubes of rather large diameter. The sliding gate is commonly utilized in the pumping pool or wherever the sealing interfaces allow easy installation. This includes the concrete surge tank structures at intermediate lift points where penstocks and storage tanks may be isolated by sliding gates.

In the case of intakes to the individual pumps at intermediate lift points where no pool is afforded, butterfly, sluice or needle valves might be used.

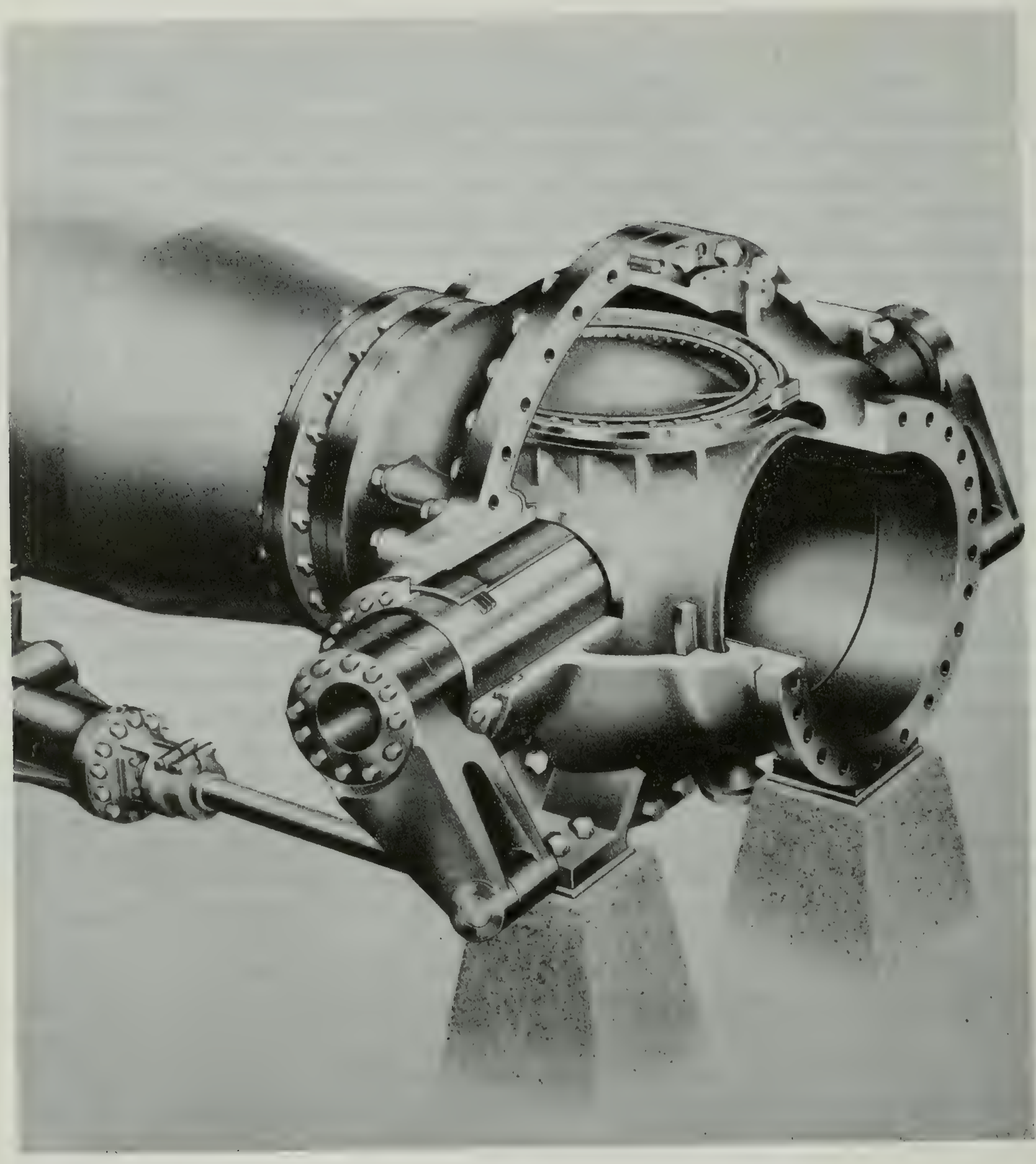


FIG. 5-1

TYPICAL LARGE SPHERICAL VALVE

It is the opinion of a manufacturer of all three that "although water tightness in the sluice valve is superior to the butterfly valves, they are large in size, particularly in height. Extra care must be taken for overall arrangement and installation, and their cost is higher. Another disadvantage to the sluice valve is that the capacity of its servomotor is large and requires an oil pressure system of particularly large capacity". The needle valve, according to the same manufacturer, is "much larger, heavier and costlier than butterfly or sluice valves. Its interior operating mechanism being perpetually immersed in water, rust caused by impurities and slime often disrupts the smooth action after operating for many years".

Discharge valves for the lift concepts must operate under heads ranging from about 650 feet for the three-lift to about 1,950 feet for the single-lift. The 650 feet head associated with the three-lift is approximately the upper limit of head for which butterfly valves are generally selected according to a manufacturer of all common types. The rotary valve, on the other hand, has an applicable range of pressure which is usually from 150 to 3,000 feet and over in water column. It is widely used with medium and high head water turbines as an inlet valve and with pumps in the same pressure range as a discharge valve.

A maker of cone type rotary valves states that they "have greater initial shutoff than either gate or butterfly valves. Closing time can be closely controlled". With regard to the relative merits of ball and cone rotary valves, this manufacturer states that the ball valve "has slightly greater head loss than cone valve", but also states that "ball valves will give many of the advantages of cone valves, and are less expensive". The statement continues, "closure of ball valves is quick and easy, with an initial shutoff almost as great as that found in cone valves".

A spherical valve manufacturer claimed the following advantages for the spherical or ball valve over the cone:

1. Reliability

Of primary consideration is the ability of a valve to dependably close, open or throttle as required, after extended periods of operation. One cause of failure to function properly is brought about by abrasive material or other foreign material which accumulates in the valve. The cone valve is more susceptible to scoring of the seats and failure to properly seat due to the conical wedging action of the plug which must be moved downward into a cavity for an effective seal. This cavity is at the bottom of the valve and susceptible to accumulation of foreign material. In addition, the lower bearing trunnion is located at the bottom of the valve making the hazard of foreign material entering the bearing surfaces greater.

The sphere valve is installed with the valve trunnion in the horizontal plane reducing the hazard of foreign material entering the bearing surfaces. When seating, the motion of the plug seat unto the body seat is that of a wiping action which tends to clear the seat of foreign material.

Simplicity of the actuator mechanism of the sphere valve favors reliability in that only one direction of motion is required, whereas both a rotary and downward motion must be imparted to the plug of a cone valve.

2. Maintenance

The general construction of a cone valve consists of a one-piece body, plug and head cover. The valve body is designed such that the plug can be inserted through the top. The head cover seals the assembly port for the plug and acts as the bearing for the plug trunnion. In order to fully inspect the seating surfaces of the cone valve, the head cover and plug must be removed. The diameter of the head cover must be larger than the normal diameter of the valve making dismantling time-consuming and requiring of heavy lifting equipment.

The general construction of this sphere valve consists of two body halves, a plug and two companion flanges. The companion flanges contain the body seats. The valve seats can be fully inspected by removal of the companion flange, which requires much less time and equipment than removal of the head cover and plug of a cone valve.

The main bearings of a sphere valve, being in the horizontal position, allow greater accessibility for replacement of bearings.

3. Space and Weight

Due to the symmetrical configuration of the sphere valve, less space is required. The cone valve must be provided with a large diameter head cover which increases the face to face width and height dimensions beyond those of a comparative sphere valve of this same nominal diameter.

Correspondingly, the total weight of the valve increases with the size. In particular, the weight of a cone valve for high head application is considerably greater than for a sphere valve.

4. Special Features

A sphere valve can be equipped with movable body seats which greatly increase the length of the seat's life by providing a controlled seating pressure and eliminating rubbing action when the valve seats. This is particularly advantageous when high pressures are involved as the loading on the plug is high. Movable seats also provide for a take-up due to normal seat wear and increases of bearing clearances.

When movable seats are provided on both the upstream and downstream side the downstream seat is used for normal operation. In the event of a rupture of the downstream seat the upstream seat is available, providing in a sense two valves in one.

The dual seat also provides the advantage of inspection, refurbishing or replacing the downstream seat without de-watering the line or fully dismantling the valve.

5. Economy

Cone valves of certain size and pressure, where the quantity demand is high, can be less costly than a sphere valve. This is because the cone valve has fewer parts and the savings in labor offset the higher costs for material. For non-standard or engineered valves the sphere valve is more economical due to the symmetrical design and savings on weight. These savings increase greatly as the size and pressure increase.

E. OPERATION AND MAINTENANCE:

A study of 113 generating plants operating for a one-year period revealed that repair and inspection of intake and head gates amounted to 1.6 percent of the total maintenance outage for the year. Surveys of maintenance experience at large pumping plants throughout the United States and Europe revealed that valves and gates are generally quite trouble-free in comparison with pumps, motors, switchgear, and bearings.

Some galling and undue wear in the operating mechanisms for cone type rotary discharge valves at Intake, Gene and Iron Mt. pumping plants

was reported in 1942 after about two years of operation. A bronze face was put on the rotators and they have since operated in a satisfactory manner. A manufacturer of these type valves claims they have been in service for thirty years without replacement of seats.

An American manufacturer recommending the double-seated sphere pump discharge valve with movable body seats states that replacement or repair of the upstream (pump side) seats can be accomplished without dewatering the discharge pipe. This manufacturer reports that their spherical valves have been highly dependable and trouble-free on high head Francis and Pelton type turbines over the past 20 years. Another recommendation is that the valve be equipped with an operator sequencing mechanism to insure that body seats are open prior to plug rotation. The only report of a field problem was due to improper operation and the sequencing devices remedied this. This manufacturer also recommends that the upstream and downstream seat rings be physically inspected annually for wear and damage from foreign objects. The valve and operator mechanism should be lubricated monthly, depending upon usage, as a minimum.

The following spare parts inventory should be kept on hand for pump discharge valve maintenance:

1. Upstream movable stainless steel type 316 seat rings with seals (approximately 1 per 2 valves).

2. Trunnion shaft bearing set of bronze sleeves and bushings (approximately 1 set per 2 valves).

3. Square plaited braid flax or asbestos lubricated with graphite packing for trunnions (approximately 1 set per 2 valves).

4. Packing for operating cylinder (approximately 1 set per 2 valves).

5. Gaskets for erection coupling (approximately 1 set per valve).

6. Gaskets for downstream companion flange (approximately 1 set per valve.

7. Spare oil pump for hydraulic system.

8. Spare bearings for oil pump.

9. Spare bearings for air compressor.

10. Spare drive belts for air compressor.

11. Spare automatic sequencing unit for hydraulic system.

A European manufacturer specializing in design and manufacturing of shut-off valves reports that their double seated spherical valve is so designed that all parts of the valve which are subject to wear, may be overhauled while the discharge line is under pressure. These parts include service seats, packings of the hydraulic drive and trunnions, control units, and by-pass assemblies.

One American manufacturer has designed a double-seated spherical valve in which the seats are energized by penstock water pressure instead of oil pressure which is used to operate the movable seats on the valves discussed previously. Stainless steel seat rings are constructed with bolted-in retainer segments which may be removed from inside the pump discharge line leading to the valve to replace a one-piece urethane sealing member. This seal replacement may be done with 1100 psi differential across the emergency seat. This valve design is notable in that it is not necessary to disconnect the pipe between the pump and the valve in order to replace the "normal seat" seal. The emergency seat could be supplied with either internally-replaceable parts or non-replaceable seats.

The sphere (plug) may be coated at the seat contact surfaces with a hard metal facing of a corrosion resistant material to provide greater life, and further coated with a baked-on polytetrafluoroethylene to provide corrosion resistance and a lower coefficient of friction on the rubbing surfaces.

Trunnion stem seal replacement is accomplished by removing the operating mechanism stem lever to gain access to a bolted plate over the stem and removing a one-piece seal ring containing grooves for a dynamic seal and a static seal of Buna-N rubber. This is performed with the valve closed and the "emergency seat" energized. The "emergency seat" is energized by closing two by-pass valves and opening a drain valve.

Normal seat seal replacement requires dewatering the line between the valve and pump, removal of locking segments. Seal retaining segments

can then be retracted, permitting good access to the mechanically locked-in urethane seal insert.

Valve operator lubrication fittings would be provided at all pin joints on the cylinder and torque arm assembly. The rod packing on the cylinder is contained in a cartridge which simplifies repacking if it should become necessary. The hydraulic power package and accumulator should be regularly inspected to see that the filters and strainers are clean, oil level is adequate, and that all valves including relief valves are in proper adjustment and working order.

The pump inlet butterfly valves for the upper plants of the two-lift and three-lift concepts can have either rubber or metal seats. One American manufacturer states that a rubber seated valve would have the lowest initial first cost. They recommend, however, that one field replacement of a rubber seat would pay for the additional first costs of a metal seated valve. They propose to make the body and seat rings of cast bronze.

Another American manufacturer states that their construction experience with high pressure butterfly valves dates back to year 1910. Their design is based upon conservative factors of safety and safe working stresses to guard against damaging deflections and fatigue strain.

Still another American manufacturer suggests a hard-rubber seat ring opposite the valve disc seal ring. They also recommend cast body sections, stating that the seals would not be effective if light fabricated sections are used.

A European valve manufacturer has an inflatable rubber hose packing which is fixed to the valve body with stainless steel sections. Leakage losses with this type of seal are no longer influenced by the deformation of the disc and are practically negligible. The specially profiled hose can be replaced without dismantling either the valve or the operating mechanism. In the closed position of the disc, the rubber hose is inflated by air pressure of six (6) atmospheres pressing it against the disc circumference. A filler ring inside the hose prevents the hose from collapsing under the valve service pressure.

Opening and closing time for the pump discharge valves must take into account the effects of water hammer. The first 80% of valve closure restricts the flow through the valve much less than 80%, because the pump pressure increases with decreased flow to provide the pressure drop required to move the slightly reduced flow through the partically closed valve. The pump characteristic curves, when available, can be used to establish a precise relationship. The decrease in velocity of fluid flowing in a line sets up a pressure wave that travels at the speed of sound through the fluid. The change in velocity of fluid flow while the initial pressure wave is propagated outward and is reflected back from the valve to the inlet or open end of the discharge pipe. For the single-lift concept with a penstock approximately 8250 feet long, the shock will travel from the valve to the top of the lift and return in approximately 3.5 seconds. Shutoff is equivalent to instantaneous when the time of closure does not exceed that required for the wave to travel from the valve to the extreme end of the penstock and back to the valve. Permissible velocity change for 25% surge, based on normal pumping velocity in the penstock of 12.4 ft./sec. is approximately 3.4 feet/sec. The deceleration is therefore 3.4 ft./sec. in 3.5 seconds or 0.97 ft./sec. The most rapid valve closure time for stopping the flow from 12.4 ft/sec. is 13 seconds for a 25% maximum surge pressure. Shorter closure periods would produce greater surge pressures. This pressure would be produced if all valves closed simulataneously.

The shortest permissible closure time of 13 seconds for the single-lift concept would require constant deceleration of water flow which means rapid valve closure initially and decreasing rate of closure as the closed position is approached. Rapid closure of the valve to 20% port opening can fall within a 10 to 20 second period and be acceptable from the standpoint of water hammer. Closing the port opening the final 20% is even more effective in decelerating the flow of water than the first 80% of opening and, hence, a source of water hammer. Closing the last 20% in approximately 20 seconds should hold water hammer to within reasonable limits and adjustment of from 20 to 40 seconds for the last 20% of port opening provides the capability for fairly rapid but safe closure speeds.

The two lift and three-lift concepts contemplated have shorter penstocks between pump stations and from the upper pump station to the top of the lift. The discharge pressures are lower and 25% overpressure is less for the multi-lift concepts. The net effect is that required closure times are essentially the same for all lifts.

F. COST - WEIGHT RELATIONSHIPS:

The cost of spherical shutoff valves provided by potential suppliers of the valves are shown in the following Table for each of the lift concepts studied. Development costs and total cost of valves for the completed pumping plant or plants is shown. The development cost will be applied to the first order where construction is performed in stages:

TABLE 5-I - VALVE COSTS AND WEIGHTS:

One-lift Pump Discharge Valves - Rotary

Diam Inches	Cost	16 Valves	Development Cost	Total Cost Discharge Valves	Weight per Valve
48.5 I. D.	$160,000	$2,560,000	$40,000	$2,600,000	125,000 lb.
52 I. D.	180,000	2,880,000	40,000	2,920,000	150,000

Two-lift Pump Discharge Valves Rotary

		18 Valves			
63 I. D.	$180,000	$3,240,000	$40,000	$3,280,000	125,000 lb.
66 I. D.	200,000	3,600,000	40,000	3,640,000	150,000

Three-lift Pump Discharge Valves - Rotary

		27 Valves			
60 I. D.	$150,000	$4,050,000	$40,000	$4,090,000	120,000 lb.
63 I. D.	170,000	4,590,000	40,000	4,630,000	135,000

Multi-lift Inlet Valves - Butterfly

		Development Cost	Two-lift 9 Valves	Three-lift 18 Valves	Weight per Valve
102 O. D.	$ 60,000	$40,000	$580,000	$1,120,000	40,000 lb.

CHAPTER 6

HYDRAULIC TRANSIENTS STUDY

A. PURPOSE AND SCOPE

The successful operation of a pumping plant can be insured only by taking due consideration of transient conditions. The planned starting and shutdown of pumps must be correctly controlled and proper provision must be made for conditions of power failure, emergency pump shut-down, etc. The behavior of the pump, drive motor, and discharge valve are the essential features of transient operation and they must perform in proper relation to the conditions imposed by the penstock, surge tanks, etc. Study of hydraulic transients is an essential part of pump selection and is part of the DMJM research study for Tehachapi pumps.

The current study is intended to give sufficient comparative information about the lift concepts to aid in that choice. A final study will be made when a particular lift concept is selected. In anticipation of final studies, model test firms are to perform three-quadrant pump testing, thrust tests, and vibration tests as a part of their programs. For the preliminary hydraulic transient study, estimates of pump performance and physical information about prototype pumps was sought. It was expected that S. Logan Kerr, DMJM consultant and recognized expert in water hammer problems, would perform preliminary computations. However, it was learned that preliminary calculations had been made by State engineers as part of their systems studies and these calculations were reported in Section M-6 of the Department's September 1964 report, "Preliminary Report of Technical and Economic Feasibility of Single-Lift, Two-Lift and Three-Lift Systems; Tehachapi Pumping Plant." As a consequence, Mr. Kerr was asked to simply review the water hammer studies performed by the Department of Water Resources. It is to be noted that his work concurs with that of the State and no further studies will be made until the lift concept has been chosen. Mr. Kerr's comments follow:

1. In order to evaluate Appendix M-6, "Water Hammer and Surge Studies," I selected the single-lift proposal and carried out a graphical analysis using the Bergeron procedures. Prints of the various diagrams utilized are:

SK 1442/1 - Surge study for single-lift alternative, assuming that sufficient pumps were in operation at the time of power failure.

SK 1442/2 - Pressure time curve covering the initial down surge only.

SK 1442/3 - The velocity time curve in an equivalent single conduit.

SK 1442/4 - The pump speed versus time immediately following a power failure.

SK 1442/5 - The approximate profile and estimated down surge gradient.

2. The maximum and minimum pressures, and in fact all of the pressures, are shown in actual units of feet head of water rather than the percentage of normal head as was included in the (DWR) report and follows the practice of Mr. Parmakian.

3. The results agree with a similar case in the DWR report within the limits of reasonable accuracy. It is necessary, however, to point out that the down surge gradients as shown in the DWR report for all of the schemes and of the case which we examined are dependent on having an extremely heavy flywheel effect not less than two million pounds feet squared. If a lower value is used for the combined motor and pump, the shape of the down surge gradient will be altered substantially and may exceed the barometric limit with a consequent parting and rejoining of the water column.

   On the sketches mentioned above that show the graphical analysis for Case I (single-life) with a flywheel effect of about two million, we have added data giving the estimated conditions for an extremely low value of flywheel effect.

On SK 1442/1 we showed the estimated down surge and up surge if there was an extremely low value of flywheel effect in the pump and motor.

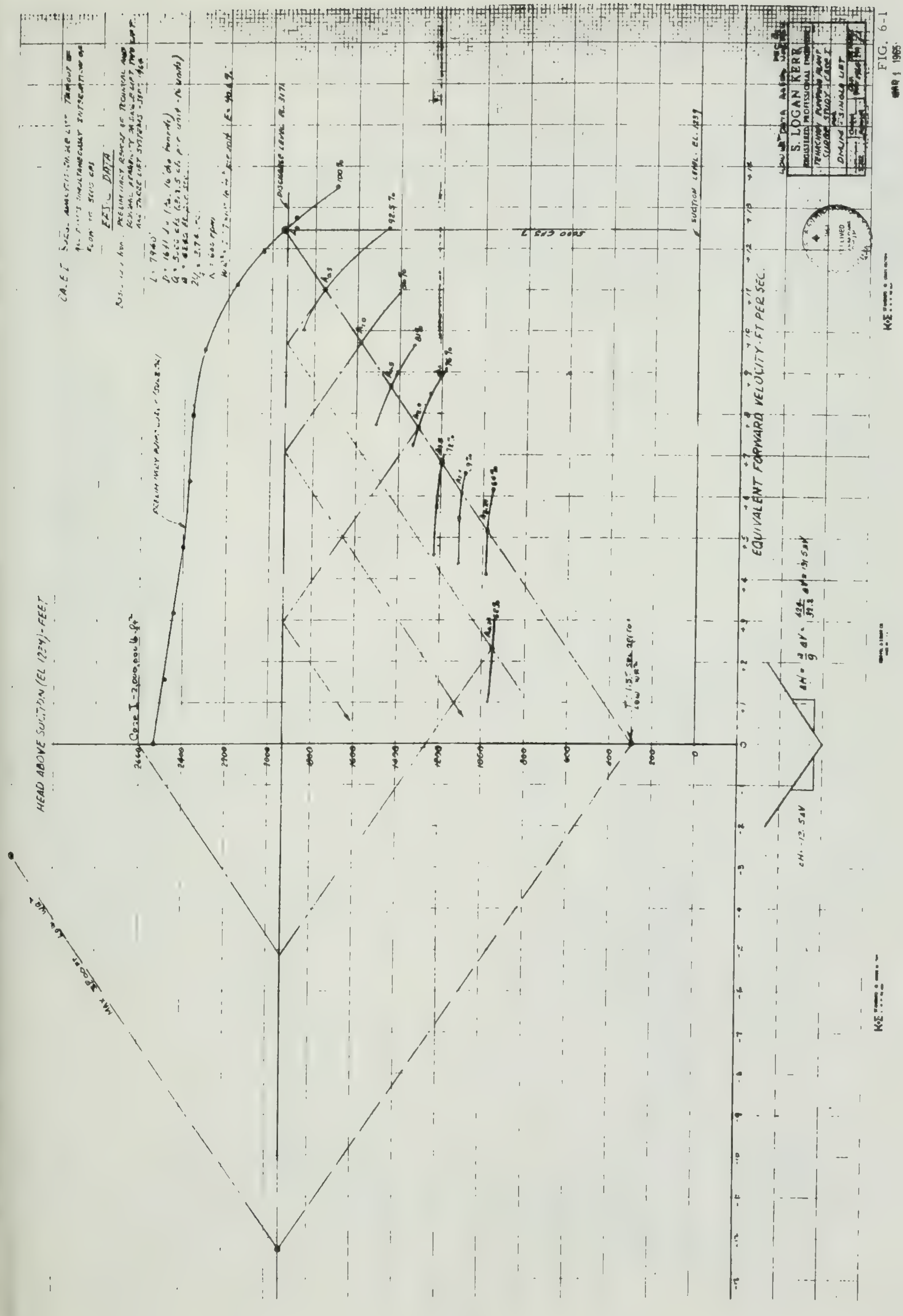
We have extended this to the other diagrams and have estimated that the rundown time would be on the order of 1.55 seconds and the water column would come to rest in about that time with a down-surge equal to 300 feet above the suction level, or approximately elevation of 1539 feet. It is also assumed that the check valve closure would be at the point of zero flow. We did not attempt to incorporate the reverse flow through the pump with the check valve open; however, this could be calculated if we know the reverse characteristics of the pumping units. We could estimate them from information available on other installations or tests, but for the present we felt that the demonstration was sufficient as an approximation of the down-surge gradient with a small flywheel effect.

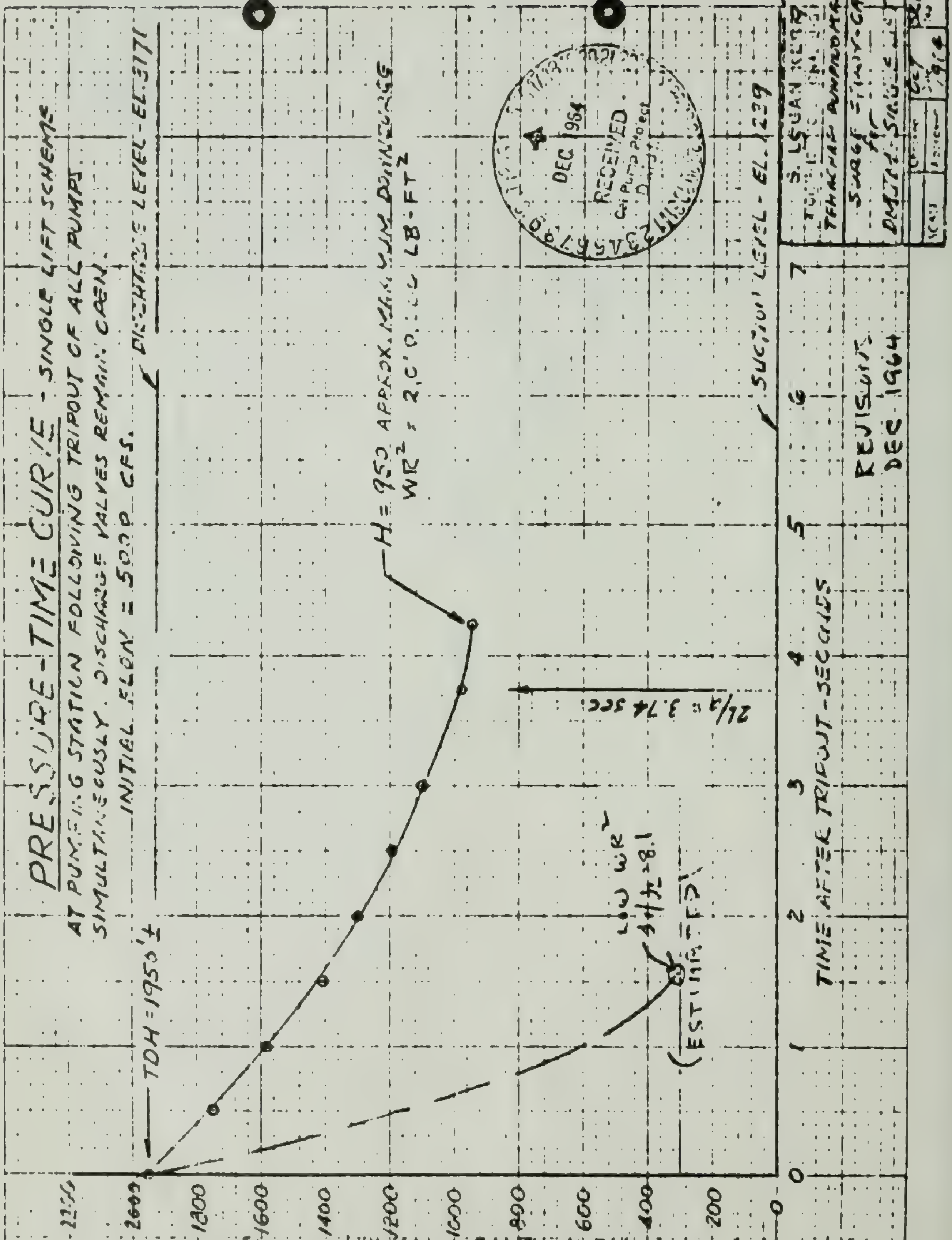
On SK 1442/5 we have indicated an approximate down surge gradient whereby the negative wave would travel undiminished to approximately Station 77 or about 3500 feet from the point of relief to the outlet of the pump discharge conduits. It should be noted that, utilizing the estimated pipe line profile from DWR Drawing M6-1, the conduits would be under a vacuum from Station 67 to approximately Station 90.

4. Similar characteristics of a less severe nature would result from a similar analysis on the two-lift and three-lift schemes. It is, therefore, highly desirable to secure the results of the Model Tests with their complete reverse flow characteristics before a decision is reached as to the profile of the conduits.

5. In the two-lift and three-lift schemes the balancing tanks are located at some distance from the discharge pipe line which, in my opinion, can have an adverse effect on their functioning. On several projects where I have been a consultant we have utilized balancing tanks on a straight

through basis and found that they functioned exactly as the design required. There are many factors relating to balancing tanks that affect their design in relation to any particular application, and at this time I have not attempted to make an exhaustive study of their use at Tehachapi. I can advise, however, that on at least three major projects they have been most successful, and have eliminated elaborate control systems and when placed in service have functioned without difficulty. I would suggest that prior to the decision concerning the 1, 2, or 3 lift scheme that such a study be made.


HEAD ABOVE SUCTION (EL 1234) - FEET
EQUIVALENT FORWARD VELOCITY - FT PER SEC.
DISCHARGE LEVEL
SUCTION LEVEL
S. LOGAN KERR
REGISTERED PROFESSIONAL ENGINEER
FIG. 6-1


PRESSURE-TIME CURVE - SINGLE LIFT SCHEME
AT PUMPING STATION FOLLOWING TRIPOUT OF ALL PUMPS
SIMULTANEOUSLY. DISCHARGE VALVES REMAIN OPEN.
INITIAL FLOW = 5000 CFS.
TDH = 1950'±
DISCHARGE LEVEL - EL. 3171
H = 950 APPROX. MAXIMUM DOWNSURGE
WR^2 = 2,000,000 LB-FT^2
LOW WR^2
(ESTIMATED)
2L/a = 3.74 sec.
SUCTION LEVEL - EL. 1239
TIME AFTER TRIPOUT - SECONDS
HEAD ABOVE ... (EL. 1239) - FT.
0
200
400
600
800
1000
1200
1400
1600
1800
2000
0
1
2
3
4
5
6
7
REVISIONS
DEC 1964
S. LOGAN KERR
TEHACHAPI PUMPING PLANT
SURGE STUDY - CASE I
for
DEC 1964
RECEIVED
FIG. 6-2

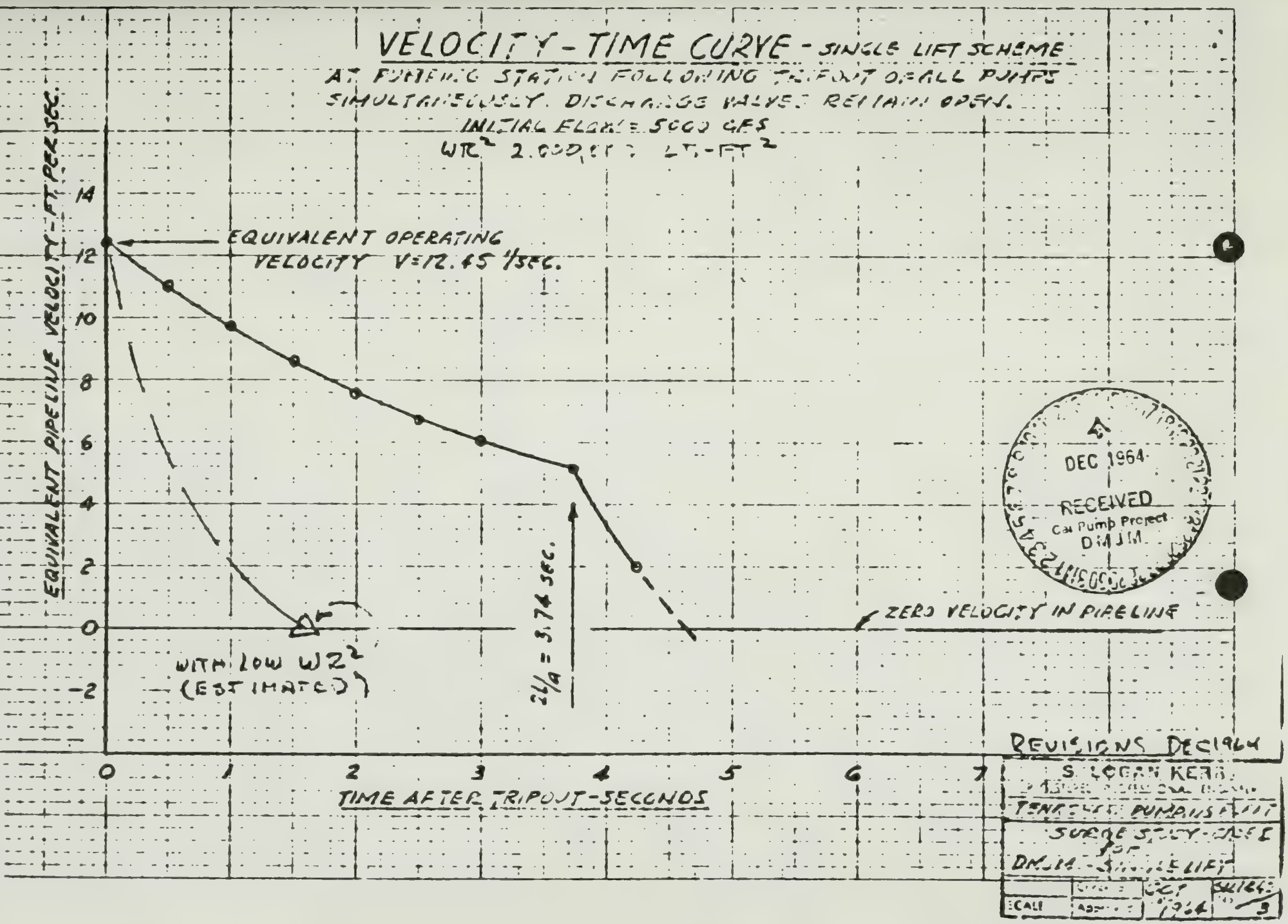

VELOCITY - TIME CURVE - SINGLE LIFT SCHEME
SIMULTANEOUSLY. DISCHARGE VALVES REMAIN OPEN.
EQUIVALENT PIPELINE VELOCITY - FT. PER SEC.
EQUIVALENT OPERATING VELOCITY V=12.45 '/SEC.
WITH LOW WR^2 (ESTIMATED)
$2L/a$ = 3.74 SEC.
ZERO VELOCITY IN PIPELINE
TIME AFTER TRIPOUT - SECONDS
DEC 1964
RECEIVED
Cal Pump Project
DMJM
REVISIONS DEC 1964
S. LOGAN KERR
SURGE STUDY - CASE I

FIG. 6-3

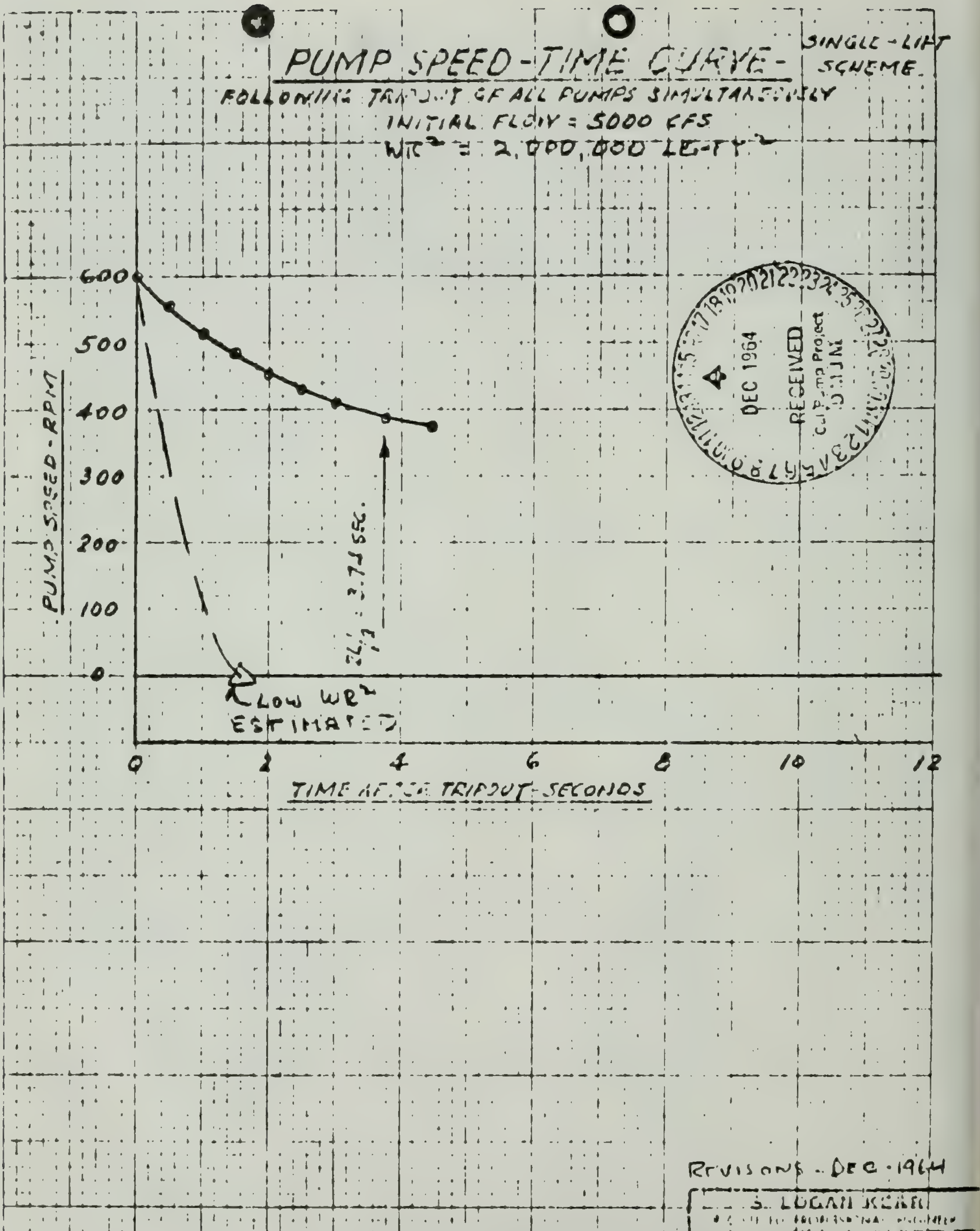
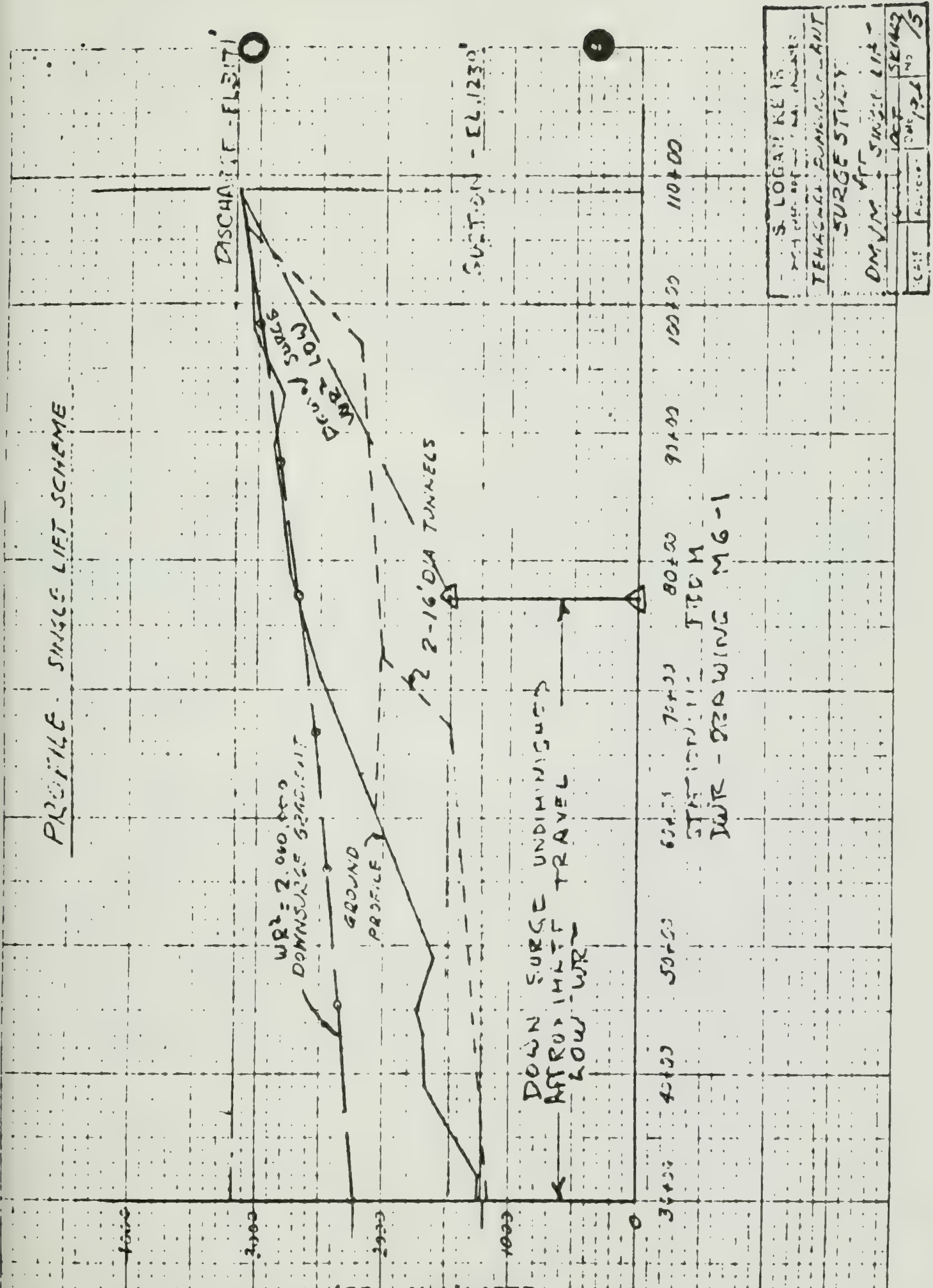

PUMP SPEED-TIME CURVE-
SINGLE-LIFT SCHEME
FOLLOWING TRIPOUT OF ALL PUMPS SIMULTANEOUSLY
INITIAL FLOW = 5000 CFS
WR² = 2,000,000 LB-FT²
PUMP SPEED-RPM
600
500
400
300
200
100
0
LOW WR² ESTIMATED
0
2
4
6
8
10
12
TIME AFTER TRIPOUT-SECONDS
DEC 1964
RECEIVED
REVISONS - DEC - 1964
S. LOGAN KERR
SURGE STUDY - CASE I
SINGLE LIFT

FIG. 6-4



PROFILE SINGLE LIFT SCHEME
DISCHARGE - EL. 2171'
SUCTION - EL. 1230'
2-16' DIA TUNNELS
GROUND PROFILE
DOWN SURGE UNDIMINISHED
APPROXIMATE TRAVEL
LOW WR²
ELEVATION - FEET
2000
1000
0
110+00
80+00
S. LOGAN KERR
TEHACHAPI PUMPING PLANT
SURGE STUDY

FIG. 6-5

CHAPTER 7

RELIABILITY STUDIES

A. PURPOSE AND SCOPE

The reliability study conducted for the Tehachapi Pump Program has three main purposes: (1) predict reliability and availability of the single-lift, two-lift, and three-lift concepts; (2) provide a basis for optimizing designs of the pumping systems under consideration by revealing those materials and configurations which will function in the most trouble-free manner; (3) provide information for spare parts ordering and maintenance planning.

Reliability and availability predictions as presented in this interim report should be qualified by stating that they are valid only as indications of the comparative standings of the reliability for the lift concepts. Absolute reliability and availability values will be derived on completion of statistical repairability studies. These values will better represent relative concept ratings but preliminary calculations indicate no changes in rank will result. Interim predictions are valid on a comparative basis because they reflect frequency and duration of unscheduled outages which are the overriding constraints on series operational modes as planned for the two and three lift Tehachapi concepts.

The scope of the reliability study is restricted to pumps and pump components as major considerations with accessory equipment such as motors, valves, bearings, and so forth receiving minor attention. It is also limited to Tehachapi operations as isolated from those of upstream and downstream plants. However, detailed technical analysis of the pumps as they affect reliability and availability will yield the most valid predictions for the lift concepts since pump maintenance is singularly amenable to accurate weighting and prognostication. It follows that pump component design and material selection as guided by results of this study will do more to enhance operational success at Tehachapi than any other planning function. It is also apparent that maintenance and spares planning will be most effective if its most accurately predictable facets receive greatest stress.

B. SUMMARY AND CONCLUSIONS

The three main purposes of the reliability study have been accomplished to a limited extent for this interim report. Results based on the studies conducted thus far are given below along with the major qualifications which must be considered in their use or interpretation. A summary is also given of the lift concept differences and the study approach which let to the interim results. A functional block diagram is given in Figure 7-1.

Results to be reported on the progress of the reliability study and qualifications of the results are:

1. Reliability, Availability, and Concept Effectiveness

The following table presents preliminary results of the reliability study suitable for comparing the three lift concepts.

Pump Concept	Concept Effectiveness	Operational Reliability	Operational Readiness
Single Lift	.979	.997	.982
Two Lift	.950	.970	.980
Three Lift	.902	.921	.979

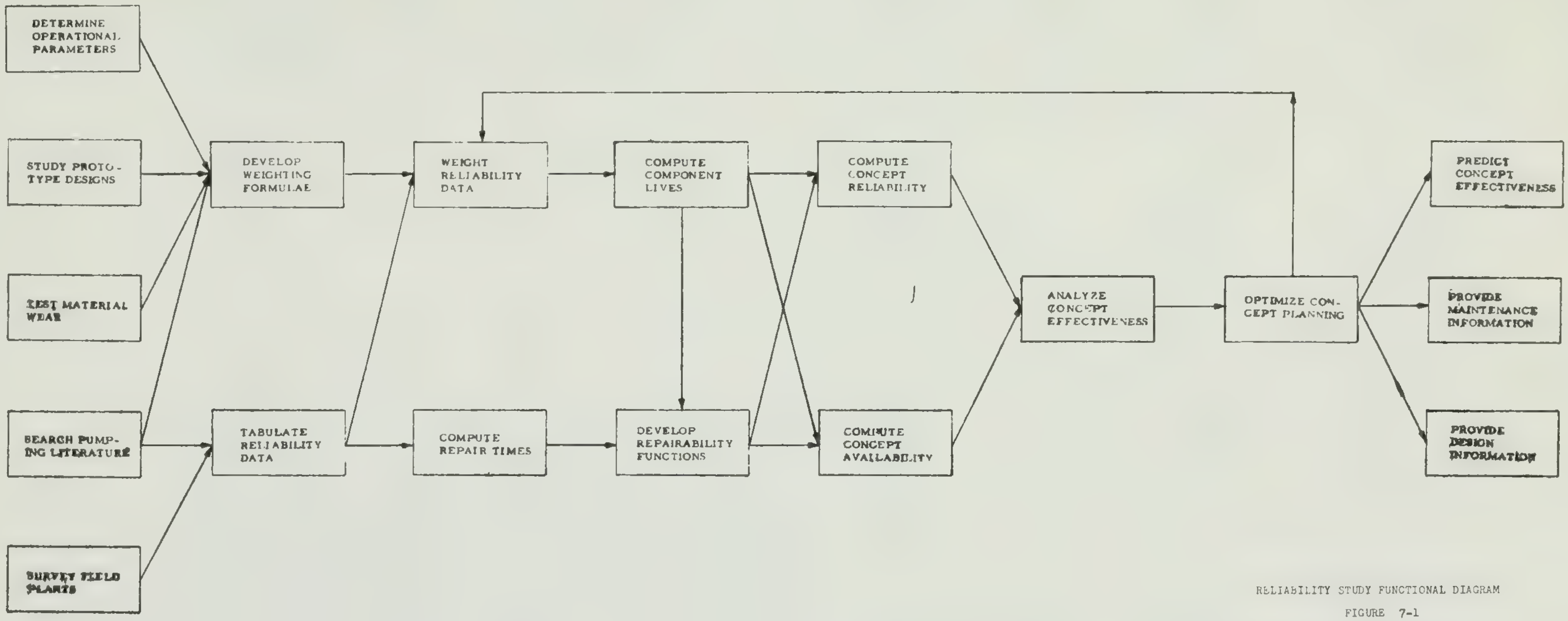
Chief qualifications of the comparative predictions are as follows:

a. Only maintenance of worn pump components is considered "scheduled" outage.

b. It is assumed that all worn pump component maintenance is accomplished in the "scheduled" repair time and consequently the frequency of such outage is not included in mean time between outages.

c. All outages for maintenance other than those connected with worn pump components are considered to be "unscheduled" but the total percentage of time taken per year for "unscheduled" outage per plant is considered to be the figure derived by the Department of Water Resources (Ref. 11) for the these lift concept and weighted for the one and two lift.

d. The mean time between "unscheduled" outages (as defined in "c" above) is taken from incomplete maintenance records of five similar pumping plants.



RELIABILITY STUDY FUNCTIONAL DIAGRAM

FIGURE 7-1

e. Mean time of each "unscheduled" repair is based on the percentage annual outage time and mean time between outages (as described in "c." and "d." above), and the number of repairs allowed annually is considered to be this mean time per unscheduled repair divided into the free time in the year aside from needed operating time and annual mean "scheduled" repair time.

f. Annual mean "scheduled" repair time is based in part on time at which the lift concept prototype components have reached wear limits based on fixed percentages of average component clearances found in the field survey. This puts these times on a physical wear basis rather than an economic one.

g. Effects of upstream plant outages on Tehachapi operations are not considered.

h. Preliminary calculations of lift concept reliability using procedures which attempt to remove all of the above qualifications (as outlined in Section F. 7) indicate that while absolute reliability figures will not agree with those given here, the comparative order of availability, reliability, and effectiveness figures for the lift concepts will not be altered.

2. <u>Design Recommendations</u>

With regard to providing a basis for optimizing pumping system design, a limited number of results may be reported. It appears, first of all, from the results of the field survey and weighting process, that water quality is the dominant factor in determining component life and that solid particle abrasion is the chief wear mechanism to be considered. Therefore, abrasion resistance should rate high on the list of qualities for material selection. Secondly, it appears that water velocity has almost as great an influence on wear as water quality and that much can be done in the field of proper component design to control this factor. However, there are common sense limits of workable clearances, space limitations and expense which must be considered in component design. Thirdly, it may be reported that the average water delivery plant considers satisfaction of water user's demands first and costs of repair second when determining worn component replacement times. Therefore, component designs and materials should be chosen on the basis of which afford the greatest probability of satisfying annual demands and on efficiency-loss costs versus replacement costs; in other words, on a cost-effectiveness basis.

The above results must be qualified by stating that they are based on a limited number of controlled experiments and on rather broad interpretation of field survey results. They also reflect a rather narrow outlook on pumping "system" optimization in that they consider only pumps and pump components whose chief mode of failure is wear.

3. Maintenance Planning

The study has provided the necessary analytical tools and formulas for predicting mean times between replacement of worn pump components and therefore provides a basis for spare parts ordering and maintenance planning. The preliminary prediction is for replacement of fixed metal bushing shaft packing every ten to fifteen thousand operating hours and general overhaul about every twenty-five to fifty thousand operating hours. The overhaul would include replacement of interstage seals and wear rings and repair or replacement of impellers and guide vanes, if any. The study also revealed that overhaul time for a multistage pump would probably exceed that for a single stage pump and that balance labyrinth overhaul about every twenty thousand hours would have to be added to other maintenance actions for the single lift prototype pump.

Qualifications for the maintenance planning outlined above are chiefly those of predicting what the water quality at Tehachapi may be like and of incomplete analyses on the tolerances which should initiate part replacement action. Also it is assumed that the installation will be free of cavitation damage. Again the results are limited to pump components.

4. Specific Observations

The foregoing results are of a general nature with regard to the reliability study as a whole. The following specific observations on lift concept traits were made in the course of the study.

a. Identical pump components will wear at a rate which is approximately proportional to a power of relative water velocity.

b. Wear in the suction impeller of the Tehachapi four stage prototype pump should proceed at a slightly faster rate than in the two stage which, in turn, should proceed at a slightly faster rate than in the single stage because of relative inlet water velocities.

c. In the absence of defined cavitation, repair of worn suction impellers is generally scheduled to coincide with overhaul time.

d. In view of the relative similarity of impeller wear rates for the three prototypes and design objectives which promise adequate submergence limits, it is assumed that impeller repairs coincide with pump overhauls in each lift concept.

e. It is found that multistage, single suction pumps generally suffer outages in addition to those for overhaul in order that balancing labyrinths may be replaced.

f. Because of balance labyrinth repair it appears that the single lift prototype pumps will experience the highest frequency of scheduled outages.

g. Surveys reveal that average overhaul time for a multistage pump is greater than that for a single stage pump because of a greater number of stages which must be dismantled and a greater number of components which must be maintained.

h. Wear rings with identical leakage characteristics subjected to single stage operational parameters in the Tehachapi three lift concept should wear at a faster rate than those in the other pump concepts because of higher head.

i. Time between pump overhauls is dictated by wear in the wearing rings so that overhaul outages in typical one and two lift plants should be less frequent than in a typical three lift plant used for the same application provided that identical clearance increases are tolerated.

5. Conclusion - Comparative Reliability

The traits and characteristics presented above have direct effect on the comparative reliability which is the relative chance for successful operation of the concepts. Overall comparative reliability of the Tehachapi concepts (concept effectiveness) is the product of their ability to deliver the required supplies of water (operational reliability) and the availability of pumps to operate for the needed annual time periods (operational readiness). Operational reliability is dependent on frequency and duration of outages since concept operation involves a series of interrupted operating times with maintenance action or standby time bridging the gaps. Operational

readiness is also dependent on outage frequency and duration. Application of reliability methods to the above mentioned pump analyses reveals the following:

a. Non-coincident unscheduled outages in each plant of a multi-lift concept result in increased outage time over a single lift.

b. Increased outage in multi-lift plants reduces availability and free time which may be used for repair.

c. Forced series operation of multi-lift plant configurations together with unscheduled outages results in increased frequency of outages and decreased availability of repair time per plant.

d. The culmination of unscheduled outage effects is that the single lift is more reliable than the two lift which is more reliable than the three lift concept.

6. Study Approach

A summary of the approach used to achieve results presented herein is as follows:

a. Pump maintenance and failure data along with design and operation information was gathered from a select and sizeable group of comparable pumping plants.

b. Parameters which affect pump component life were determined from theory and experience.

c. Data were tabulated according to parameters which affect component life.

d. Normalization of tabulated data was achieved through weighting factors utilizing parameters of Tehachapi lift concepts and the individual plants surveyed.

e. Reliability curves for pump components were generated from the weighted data.

f. Pump component mean lives were computed from theoretical curves best fitting the raw reliability curves.

g. Mean times for component repairs were obtained by averaging field survey estimates.

h. Component mean lives and mean repair times were used to hypothesize a planned outage schedule.

i. Mean time between unplanned outages and mean time of each outage was determined from data at hand.

j. Concept operational availability was determined utilizing mean annual planned and unplanned outage time.

k. Concept operational reliability was determined using mean time between unplanned outages, mean unplanned outage repair time and mean annual scheduled repair time using a curve drawn to estimate effects of fractional repairs on the partially redundant situation.

l. Lift concept effectiveness was computed to be the product of operational reliability and operational availability.

Prediction of absolute reliability, effectiveness and availability for the selected Tehachapi lift concepts as an isolated entity will be given in the final report after a thorough analysis of economical repair considerations and after outages for motors, valves, bearings, and so forth have been predicted on an average basis. Cost-effectiveness analyses utilizing wear test data will be instituted to optimize material selection. Design improvements will be discussed with manufacturers and maintenance actions will be planned on the basis of optimized design selections.

C. GENERAL CONSIDERATIONS

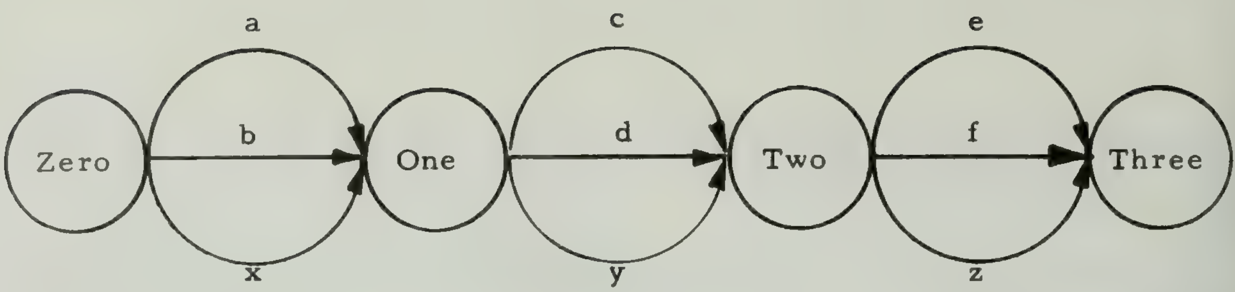
1. Reliability Concepts

Reliability is quite simply just a logical and useful method of counting. It may be imagined that Figure 7-2 represents a road map including four cities; Zero, One, Two, and Three. Highway routes a, b, c, d, e, and f between these cities are all scenic routes while x, y, and z are not scenic. If a traveler approached a fork in the road and picked any of the three routes from Zero to One at random, what are the chances that he picked a scenic route (either a or b)? The chances that a is the chosen route are one out of three. That b is chosen is also a one-third probability. Then since either of these paths would be scenic, the chances are two of three that the traveler will follow a scenic route from Zero to One. This example is illustrative of adding probabilities in an "either-or" situation (either route a or b in this case are good choices).

To determine the probability that the traveler will go all the way from Zero to Two on scenic routes it is only necessary to count up the ways in which scenic routes may be taken (a, c; a, d; b, c; b, d) and the total number of routes which might be chosen (a, c; a, d; b, c; b, d; x, c; x, d; x, y; a, y; b, y). The number of different scenic routes is 2 x 2 or four and the total number of different routes is 3 x 3 or nine. Then it is easily seen that the chances of going from Zero to Two on scenic routes are (2/3)(2/3) or 4/9. This is illustrative of multiplicative probabilities when a "both-and" situation arises, that is, the traveler had to go to both One and Two in series on scenic routes to be successful.

In the above example of counting ways to succeed, nothing was said about the time element involved in taking scenic routes. It could be that some of these routes are too long for the traveler to take certain combinations and still cover as much ground as desired in a limited vacation time. Thus another important element in reliability is time, which is also handled by counting, and it will be pointed out later that time is a factor in limiting the ways Tehachapi lift concepts may successfully deliver their water.

Continuing with the example of the traveler, it may be desirable to predict wear on the tires before making the trip so that the chances of having to replace badly worn tires before completion of the trip may be judged and preparations made. Assume the traveler has meticulously noted mileage figures at every needed purchase of a set of tires during his 25 years of driving experience. He figures that the mileage on his present set will be 20,000 by



TRIP	SCENIC ROUTES	TOTAL ROUTES			CHANCES OF TAKING SCENIC ROUTES
Zero to One	a or b	a, b, x			2/3
Zero to Two	a & c, b & c a & d, b & d	ac, ad, ay,	bc, bd, by,	xc xd xy	4/9
Zero to Three	ace, bce	ace,	bce,	xce	8/27
	ade, bde	ade,	bde,	xde	
	acf, bcf	acf,	bcf,	xcf	
	adf, bdf	adf,	bdf,	xdf	
		aye,	bye,	xye	
		ayf,	byf,	xyf	
		ayz,	byz,	xyz	
		acz,	bcz,	xcz	
		adz,	bdz,	xdz	

FIGURE 7-2 RELIABILITY THAT SCENIC ROUTES WILL BE TAKEN

the time the trip is over and he counts seven out of nine sets of tires in his past which successfully went over 20,000 miles before reaching a point of baldness so he is reasonably sure that he is safe in not making the purchase before the trip.

Of course the tire manufacturers are ten jumps ahead of the traveler and have plotted the statistical percentage of tires safely exceeding any mileage versus mileage on the basis of almost unlimited experience and controlled tests so that they know the probability of making their usual profit on a tire warranty based on tread life. They find that if a batch of tires are all tested on wear testers under controlled conditions and all treadless tires are removed from the test, then the number of tires taken out of testing every month as a percentage of those still undergoing test follows a linearly increasing function. Reliability versus mileage or time (at constant speed) is found to fit normal or Gaussian distributions very well so these distributions are commonly adopted in cases where reliability is dependent on a wear failure mode.

The traveler knows that if he had as much data on tires as the manufacturers his reliability curve for worn tires would resemble a normal curve so he assumes a best-fit normal curve for the data points he does have to enhance the accuracy of his predictions. Another technique he may use to increase accuracy is to make use of his knowledge that some of his tires ran for certain mileages without wearing out even though their final wear-out points are unknown to him due to automobile trades. He counts the number of tires surviving a certain failure mileage (including non-failures) as a percentage of the total surviving prior to the failure and the survival rate for any mile increment between failures is multiplied by the reliability at the previous failure mileage as if the tire must operate through a series of mileage intervals.

Other than using these reliability techniques, his prediction accuracy may be improved by weighting the tire-wear lives in his experience according to tire material resistance to wear (rubber versus nylon), friction factors (rough roads versus smooth), depth of tread at purchase, and speeds traveled.

Tire life is not completely determined by wear so that if a traveler wished to predict his chances of successfully completing his trip without tire trouble he would have to consider the probability of road hazards leading to a blowout. Road hazards are completely random in that time of their occurance is not predictable. If a batch of tires were observed from new until all had met with a blowout (wear being within tolerance at all times) it would be found that the number failing in any interval as a percentage of those surviving the previous failure would always be about constant. Hazard rate is constant for

random failures and the distribution which best fits this situation is the exponential as used to determine length of road hazard guarantees.

2. Reliability Definitions

It is first necessary to define reliability as it relates to the function of the Tehachapi pumping concepts. That function is to deliver water when it's needed. The function suggests that both ability to operate and availability to operate are necessary for successful accomplishment. Then Tehachapi reliability is equivalent to an index known as "system effectiveness" in reliability practice. Effectiveness is the product of operational reliability and operational readiness confirming its applicability to the "both-and" Tehachapi situation defined above. Formal definition statements for these terms are given below as found in reference 7-12*. Corollary definitions and explanations applied specifically to the Tehachapi Pumping Plant are also given where needed.

a. System Effectiveness or Lift Concept Effectiveness

The overall capability of a system is described quantitatively by the term "system effectiveness". The optimization of system effectiveness can be accomplished by a proper balance between the conflicting requirements of performance, operational readiness, and cost.

"System effectiveness is the probability that the system can successfully meet an operational demand within a given time when operated under specified conditions."

The corollary definition is: Lift concept effectiveness is a comparative index of the probability that the Tehachapi Pumping Plant can pump its yearly demand during each year from 1971 through 2040.

b. Reliability

"Reliability is the probability that a system will perform satisfactorily for at least a given period of time when used under stated conditions."

The corollary definition is: Operational reliability is the probability that a Tehachapi lift concept will operate for the duration of time needed to deliver annual water requirements.

* References are listed at end of this chapter.

c. Operational Readiness or Availability

"The operational readiness of a system or equipment is the probability that at any point in time it is either operating satisfactorily or ready to be placed in operation on demand when used under stated conditions, including stated allowable warning time. Thus, total calendar time is the basis for computation of operational readiness."

d. Redundancy

"Redundancy is the existance of one or more ways for accomplishing a task, where all ways must fail before there is an overall failure."

e. Time Divisions

(1) Operating Time

"Operating time is the time during which the system is operating in a manner acceptable to the operator."

(2) Outage Time or Repair Time

"Outage time is the total time during which the system is not in acceptable operating condition."

(3) Storage Time or Standby Time

"Storage time is time during which the system is presumed to be in operable condition, but is being held for emergency—i.e., as a spare."

3. Pump Failure Modes

The data collected in the field made it obvious that the chief failure mode in pumps is wear and that the following components are commonly replaced because of this wear:

a. Impeller wear rings

b. Interstage seal rings

c. Balance rings

d. Shaft packings

e. Impellers

Figure 7-3 gives a pictorial representation of these components along with the pump bearings which are sources of unplanned outages. The relative complexities of single stage, two stage, and four stage pumps as planned at Tehachapi may readily be seen in this figure and it is this added complexity in the multistage pumps (particularly the four stage) which makes them less reliable pumps than the single stage.

4. Pump Maintenance Practices

The following assumptions which parallel those of DWR in Reference 7-11 are made as a starting point. Scheduled outages are planned so that one unit at each plant is down for service at the same time. Maintenance work on these units is then performed simultaneously on a 24 hour basis. Maintenance personnel requirements to perform unit overhauls on this basis are being explored.

Other possibilities which can be evaluated to level out or reduce peak personnel requirements include:

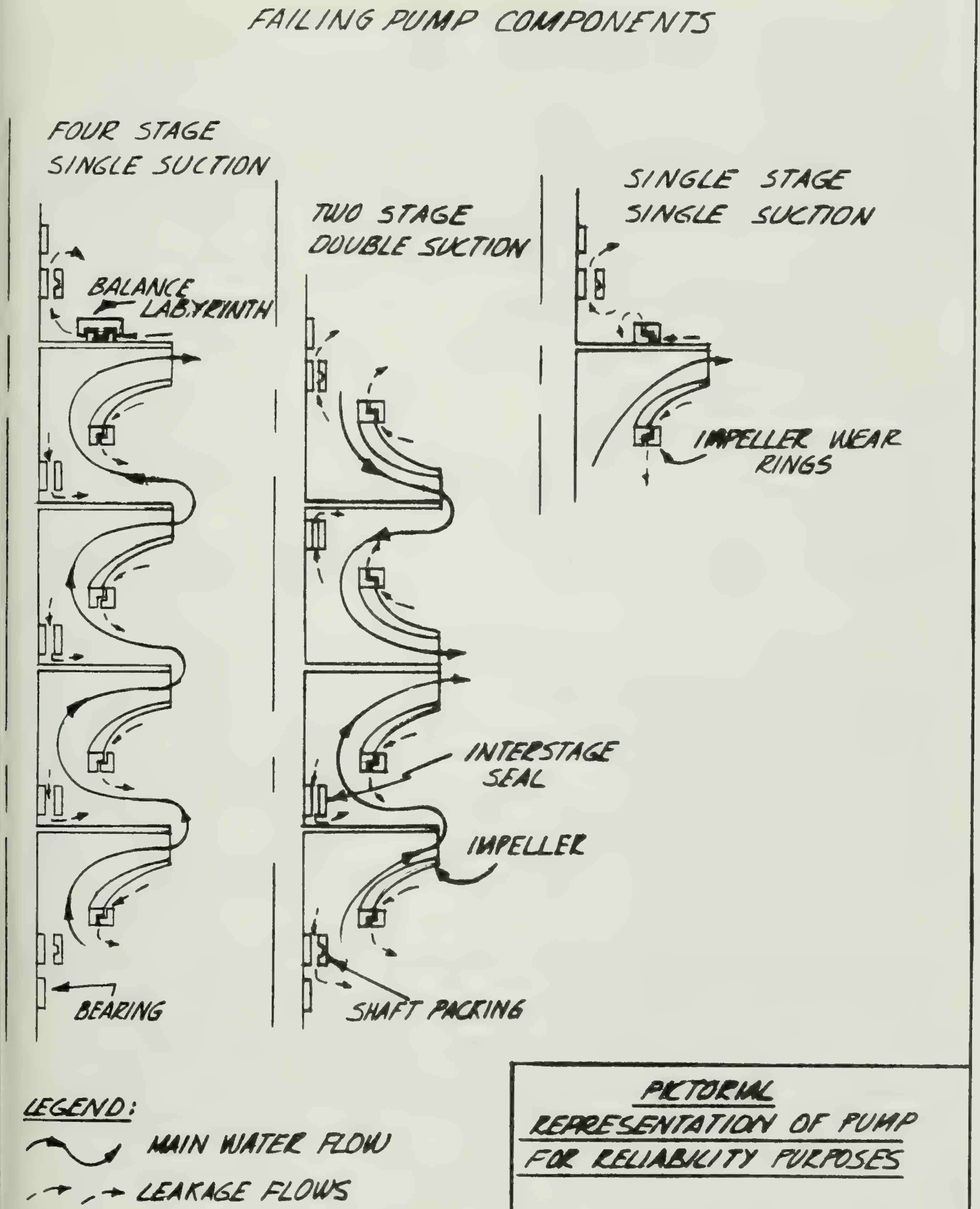
a. Sequential or partially overlapping maintenance on corresponding units of the two-lift and three-lift concepts.

b. Scheduled maintenance performed on something less than a 24 hour round the clock basis.

c. Increased plant capacity or spare units to accommodate a and b.

5. Tehachapi Operational Parameters

The assumed yearly water demand buildup and unit installation schedules are shown in Tables 6-I and 7-I of Volume I. The data presented in these tables were obtained from the basic criteria given by DWR in Reference 7-11. Based on the assumed yearly water demand and unit installation schedules, tables of accumulated unit operating hours were computed. These accumulated unit operating hours are shown in Tables 6-II and 7II of Volume I for the single-lift and multi-lift concepts respectively. The plant capacity of 5000 cfs represents 104.9% of the required continuous flow rate in year 1991 and thereafter.



PICTORIAL
REPRESENTATION OF PUMP
FOR RELIABILITY PURPOSES

FIG. 7-3

6. Tehachapi Design Parameters

Design parameters assumed for the reliability studies include the plant features given by DWR in reference 7-11 and prototype pump designs made by Allis Chalmers/Sulzer, Baldwin-Lima-Hamilton/Voith, and Byron Jackson of the 4-stage, single flow; 2-stage, double flow; and 1-stage, single flow pump types respectively. It is expected that the major impact of the reliability studies on pump design will be to point out the effects of materials selected for pump components on maintenance requirements.

7. Tehachapi Reliability Concepts

The number of ways in which a pumping plant may successfully deliver its quota of water has limits. It might operate without an outage until it has successfully delivered its quota or it might operate some length of time, suffer an outage and repair, and resume operation in time to make its water deliveries on schedule. It might also be repaired twice or three times on any number of times in the course of its operation and each of these modes of operation may result in successful delivery of water. It may be seen that the limit to the number of ways in which success is accomplished is the number of repairs which may be made in the time span of interest. Time allowed for repairs annually is idle time provided by plant overcapacity.

Two types of repairs must be made, planned and unplanned. Since the planned outages may be predicted because of their wear failure modes, these outage times are determined first and any time left after subtraction of operating time and planned outage time is time available for repair of unplanned outages. This time then must be apportioned among the lifts to make them independent of one another. That is, time available for repair of outages in the two lift concept must be divided by two because any time taken in one automatically results in the same time taken from the other even though it isn't in need of repair. In order to use the techniques outlined under Reliability Concepts, the lifts, like the routes, must be independent of each other.

As with the example of the traveler predicting wear-reliability of his tires, the reliability of the wearing pump components is based on rather limited data and several techniques are used to improve the accuracy of predictions. Data from components which have not failed is incorporated to help offset the preponderance of short times to failure necessitated by the fact that contemporary practice is of greatest interest and failures in new plants reflect unusually bad water or design conditions. Fitting a theoretical normal curve to the data points gives the distribution symmetry and also helps to avoid a reliability

skewed to the shorter times to failure which weight the study. Finally, the data lives are weighted for expected Tehachapi conditions with regard to water, design, operational, materials and tolerance parameters.

In regard to the redundancy factor used later in the text, this simply represents the counting of ways to succeed where the different ways are different numbers of repairs. To the probability that the unit succeeds for the needed time without an outage is added the probability that one failure takes place in that time and the repaired unit succeeds for the remainder of time needed. The summation used later carries this method on to any number of repairs desired where the added terms represent products of probability that each number of failures will occur and the same number of repaired units will successfully complete their period of operation.

D. STUDY METHODS

In reaching the goals of the reliability study, the basic tasks performed were the mechanics of data collection and processing, and the application of general mathematical techniques used in statistical analyses. The steps which comprise these tasks are described below. Examples of how the methods are utilized are given in Section 7. E. and tabulations of the results are given in Section 7. F.

1. Data Tabulation

A great share of the time and effort expended on the reliability study was used to gather, extract and tabulate data on field maintenance experience. Sources of the material studied included personal interviews, articles in professional journals and manufacturer's magazines, and correspondence with various plants, agencies and individual pumping problem analysts. Some forty or more pumping plants were investigated covering close to two hundred pumps operated for millions of total hours. Material of interest to the reliability study made up only a small part of the total pump research input and the needed elements had to be extracted and interpreted and interpolated to achieve a compendium of pump reliability data which is probably without precedence.

Work sheets were made up with fifty vertical columns containing elements of information, all of which were utilized in the study as will be demonstrated in later sections of this report. Plant operational characteristics were listed first including plant name, type of pumps, number and manufacturer of each, specific speed, capacity, normal submergence, total head, head per stage, speed, efficiency and starts per thousand hours. Descriptions of general water quality and water solids content followed operational parameters. Important component parameters were tabulated in all remaining columns. Included were component materials, diameters, designs, times to repair or replacement, times operated without repair or replacement, repair times, and repair remarks.

Components listed were wear rings, balance rings, interstage seal rings, packing arrangements and impellers. Design information for rings included clearances, number and average length of throttling surfaces, seal water attachments, friction coefficients, number of 90^{o} bends or pressure relief chambers and their sizes, and seal water quality. Drawings accompanied descriptions where they helped to clarify the design.

Times to failure and without failure as derived from the raw data were immediately adjusted in cases where final clearance or leakage was given. For example, a ring at Tracy Pumping Plant had worn from an average 0.037 inch clearance to an average 0.123 inches in 30,815 hours of operation. The time at which it reached the normalized standard tolerance clearance (twice original clearance) was calculated to be (30,815 x 0.037)/(0.123 - 0.037) or about 13,300 hours. This application of a judgment weighting factor is more fully explained in a later section.

Repair remarks included number of men on the crew, standard work day or week, other repairs included at that time within the same time period, and whether the pump is dismantled axially rather than horizontally.

2. Application of Weight Factors

In order to standardize varied field experiences and bring them more in line with what might have been had they occurred in the Tehachapi application, all times were weighted by various factors. The weight factors were based on common practices in pump design and maintenance as well as particular designs, operating conditions and water quality expected at Tehachapi. The derivation of these factors and examples of their application are given in later sections.

The field survey operational characteristics listed on work sheet tabulations plus their Tehachapi counterparts as found in Ref. 7-11 were used to derive water velocity weight factors which were applied to all component lives. A new listing of these weighted lives was made with plants arranged generally in order of water quality and component materials listed beside the life times. Analyses of relative component operating times for different materials but similar water were made to corroborate the choice of material weight factors made.

Material weight factors were applied to the component lives and mean lives in various water quality groups were calculated. These data were studied to confirm the choice of water quality classifications and to derive average quality weighting factors. The water factors were applied and resulting times to failure and times without failure were listed on sheets according to the component for which they were found.

3. Prediction of Component Life and Repair Times

In order to predict overall plant reliability and maintenance schedules it is necessary to predict component mean times between repairs and repair time for each component.

A mean time between repairs could be found by simple arithmetic averaging of the weighted times to repair but greater accuracy in the predicted value should be achieved if a statistical distribution for the data may be determined and the mean of this distribution taken. The reasons for this and techniques involved are outlined below.

If the rings of a pump were subjected to hundreds of controlled wear tests under conditions which were as identical as a laboratory could produce, probably no two measurements of weight lost or clearance gained would be alike to three or four significant places. If the results were examined to only one or two place accuracy a trend in the results could be seen and the re-occurrence of one or two values more than others could be discerned. In fact, if enough tests were run, the distribution of number of like results versus value of the results would appear to have a peak near the arithmetic average of the results and taper off symmetrically to either side of the peak. A smooth curve drawn to best fit the points would have a bell shape and it could be closely simulated by a normal or Gaussian distribution curve.

When examining the number of impeller or ring repairs in each ten thousand hour segment of time it is noted that some gaps appear in which no failures are recorded and other segments have far more or fewer failures in them than neighboring segments. The fewer total instances of failure reported, the more noticeable are the discrepancies. Experience has shown that these gaps and discrepancies would disappear if a sufficient number of results could be obtained. Assumption of a statistical distribution as indicated by data trends helps to smooth over these deviations from the norm and, in effect, yields results which appear to be based on many more data points than were actually available. At the same time it decreases a tendency to err in the direction of shorter times to failure caused by the fact that these times will naturally be observed more often in plants of nearly equal age.

A second reason for taking the mean of a distribution rather than averaging is that it allows for the use of operating times "without failure" as well as those "to failure." Operating data from contemporary plants and those

with very long component lives may thus be utilized. Certainly it is not wise to take 20,000 hours as mean time on the basis of ten failure observations if sixteen or twenty pumps have run this component 30,000 hours and are not yet in need of repair.

If time between repairs is taken as the random variable and probability of survival is calculated at every instance of failure, the reliability estimate becomes

$$R(t_i) = \frac{(N_1 - f_1 + 1)}{N_1 + 1} \quad \frac{(N_2 - f_2 + 1)}{N_2 + 1} \quad \frac{(N_3 - f_3 + 1)}{N_3 + 1} \cdots\cdots \frac{(N_i - f_i + 1)}{N_i + 1} \qquad (7\text{-}1)$$

where $R(t_i)$ = Reliability at time t_i.

N_i = Number of components surviving t_i including both those which later fail and those which do not.

f_i = Number of components being replaced or repaired at t_i.

t_i = Any time of interest at which planned maintenance took place.

An example of how this technique is used is given in a latter section.

A measure of how closely the estimated curve fits the normal reliability curve and of the accuracy of the estimated mean time to failure is given by the 95% confidence limits which are:

$$95\%\ \text{Confidence Limits} = (m \pm 1.96\ s/f^{1/2}) \qquad (7\text{-}2)$$

where m = estimated mean time to failure
s = standard deviation
f = the number of failure observations made. = $\sum f_i$

Standard deviation (s) and mean time between failures (m) for the best fitting normal curve may be found either graphically or as a solution to the simultaneous equations:

$$\sum_i f_i t_i = fm + s \sum_i y_i \qquad (7\text{-}3)$$

and

$$\sum_i t_i y_i = m \sum_i y_i + s \sum_i y_i^2 \qquad (7\text{-}4)$$

where y_i is the normal deviate corresponding to $R(t_i)$.

Component repair time was taken as an approximate average from repair times given by plant maintenance men. It was assumed that all repairs are made by a crew of proper size and training for the job. Whereever possible, the latest repair times were used to calculate the average since it is assumed that repair proficiency increases with the time that each plant has operated. It was assumed that field repair work was done in eight-hour days, five-day weeks, 21-day months. It was also assumed that repair times for all multistage pumps were applicable to both Tehachapi multistage prototypes and that repair times for single-stage pumps were applicable to the single-stage prototype.

4. Prediction of Planned Outage Schedule

Once component mean lives have been determined from the simultaneous equations for the best fit distribution, these mean times may be used to set up a probable repair schedule.

Mean life of the wear rings was taken as time between overhauls. Mean times between stuffing box or balance ring replacements were adjusted slightly to make multiples of these times coincide with overhaul times. This process is demonstrated later. The planned outage schedule then was plotted only to the time at which the cycle of repairs is repeated.

5. Prediction of Unplanned Outages

Unplanned outage time per year was taken from ref. 7-11 and the frequency of these outages was derived from a study of unscheduled outages for 45 pumps at five plants covering a total pumping time in excess of two million hours. Pump hours at each plant were divided by number of unscheduled outages in those hours and the resulting times between unscheduled outages were averaged to get mean time between unplanned outages.

6. Prediction of Lift Concept Availability and Reliability

Availability of a lift concept is defined as the probability that the concept is ready to operate at any time. This is called "operational readiness"

for which the symbol is "P_{or}" and it is found from

$$P_{or} = 1 - \frac{\text{(unit hours per plant scheduled outage per year)}}{\text{(8760 hours/year)(number of units per plant)}} \qquad (7\text{-}5)$$

$$- \frac{\text{(number of plants)(unit hrs/plant unscheduled outage/yr.)}}{\text{(8760 hrs/yr)(no. of units per plant)}}$$

Scheduled outage per year is calculated using needed operating time per year and mean repair time. Operating time needed to deliver 3,451,200 acre feet of water per year with 5000 cfs capacity is about 8350 hours. Predicted repair times are added up for the complete cycle of repairs and are multiplied by 8350 over the total pump hours in the cycle to get mean scheduled outage per year.

Unscheduled outage time per year is taken from ref. 7-11 as 0.31% for the three lift plant and weighted upward to 0.425% for the two lift and to 0.475% for the single lift. Weighting is based on the relative complexity of the double stage pumps in the two lift and the four stage pumps in the one lift plants.

Reliability of lift concepts is calculated on the basis of unscheduled outages only. It is assumed that planned maintenance times are chosen prudently enough on the basis of early Tehachapi experiences and safety factors so that the pumps never have to stop for replacement of worn parts before their scheduled maintenance times once continuous operations commence. Perhaps the only premature wear which could cause early stoppage is that in the packing where leakage would spill over if disposal means were not sufficient. The probability that packing will need replacement before scheduled replacement time may easily be calculated knowing the frequency distribution of the failures for all experienced conditional parameters representing limits for variance in wear which could be expected. However, this probability would be identical for each lift concept so it adds nothing to the comparison of the concepts. Another factor to consider is that early and late replacements of stuffing boxes should average out over the years as varying conditions average out.

Reliability of a pump suffering unscheduled outages is assumed to follow an exponential curve representative of a constant hazard rate. That is, since unscheduled outages are random rather than predictable, they could

come at any time and the probability of such an occurrence in any fixed interval of time is constant. Verification that reliability on the basis of unplanned outage is exponential for pumping plants was obtained by plotting it for the five plants used to determine frequency of the outages. Mean time for these reliability curves is simply the arithmetic average since no times without failure enter into it. The 95 per cent confidence limits can be described by the following equation:

$$\text{confidence limits} = \frac{2fm}{X^2_{2f}} \tag{7-6}$$

where f = the total number of failures

m = the mean time to failure

X^2_{2f} is obtained from a table of the X^2 distribution for 2f degrees of freedom at the .025 and .975 probability points.

The probability that a lift concept will operate continuously from start to "t" hours without occurrence of an unscheduled outage is

$$P_r = (e)^{-nt/m} \tag{7-7}$$

P_r = probability that the concept will run successfully for "t" hours

t = operating time of interest

n = number of plants in the lift concept

m = mean time between unplanned outages

The time of interest, "t", is chosen as needed operating time per year or 8350 hours in the case of a 5000 cfs capacity. A year's pump time is chosen, first, to coincide with the time increment normally used to calculate availability, and secondly, because it appears logical that demand for water will decrease to its lowest ebb once per year making a convenient cycle of operation.

Since the two-lift and three-lift concepts depend on pumping in series without limited storage between plants, an outage in one plant will force an equal outage in the others. The series must operate successfully in order for the concept to fulfill its requirements. Therefore, the "n" for

number of plants or lifts is included in equation (7-7).

In actual operation, when a unscheduled outage occurs, repairs will be made and the plant put back on full operational capacity. The pumps do not necessarily have to run continuously for any specified time interval such as 8350 hours. If repair can be made in allowed standby and outage time, the situation becomes identical to that in which a duplicate system is constantly on standby, ready for use when needed. The probability for successfully running 8350 hours per year with repair and restart is:

$$P_r = \left((e)^{-\frac{t}{m}} \sum_{r=o}^{k} \frac{(t/m)^r}{r!} \right)^n \qquad (7\text{-}8)$$

where k = the number of repairs and restarts allowed per year. Points were plotted for the summation above with k = 0, 1, 2, and 3, and a smooth curve was drawn connecting the points. Values of the summation were then taken off the curve when fractional numbers of repairs were allowed annually.

7. Prediction of Lift Concept Effectiveness

Lift concept effectiveness is essentially a measure of the chances that the concept can successfully deliver its predicted annual quota of water over the Tehachapis. It is the product of the probability that it can run successfully for the needed number of hours annually times the probability that it will be ready to start operating at any time of year it is needed; in other words, it is the product of operational reliability and equipment availability. The equation is:

$$P_{ce} = P_r \, P_{or} \qquad (7\text{-}9)$$

where P_{ce} = lift concept effectiveness

P_r = operational reliability (equation 7-8)

P_{or} = operational readiness or availability (see 7-5)

Lift concept effectiveness as calculated in this report is based on standard practices in pumping plants similar to those making up the concept. These practices are determined from surveys and literature. The effectiveness index is calculated for a number of over-capacity allowances.

E. QUANTITATIVE DEVELOPMENT

Once the goals, requirements and methods of the reliability study are established, it remains to perform the necessary calculations, tabulate the results and perform analyses. Computations are the primary concern of this section. Examples are given of techniques used in treating weighting factors, component lives, repair times, maintenance schedules, reliability, availability and effectiveness.

1. Derivation of Weighting Factors

The methods by which weighting factors are derived include studies of pump behavior and theory, discussions in depth with plant operators and maintenance men, and thorough examination of records, descriptions and drawings. The factors receiving earliest attention are those which are most obvious or best documented, factors about which there is little uncertainty. Once these weighting factors have been established and applied to the data it is possible to use statistical or analytical information from the data itself to get assurance that subsequent weight factors are correct or to generate weight factors without recourse to prior knowledge and experience.

A weighting study can treat a limited number of variables, therefore, assumptions are made so that selected phenomena may be deemphasized or excluded from the study. These assumptions are presented first in the following discussions and are followed by descriptive summaries of weighting factor derivations. Sample calculations and applications of the factors will be found in a following section.

a. Assumptions

The reliability study is primarily concerned with determining scheduled outages and the influence of scheduled outages on the ability of the pumping plant to deliver the required water quotas. Therefore, only predictable outages are of interest and these are limited to pump components replaced because of wear. Pump bearings were not considered to be among these components because no instance was found in which repair or replacement was due to wear. Rather the outages caused by bearings are due to high spots or seizing due to loss of oil flow which are unpredictable and better left in unscheduled outage. Scheduled replacements are made for worn wear rings, balance rings, impellers, stuffing boxes and interstage shaft seals.

A simplifying assumption was that wear or failure of any one component did not significantly affect the wear or failure of any other component. It is also assumed that the pump is initially well constructed and suffers no excess vibration. Adequate submergence of the Tehachapi pumps is assumed so that cavitation wear may be neglected. Cases in which impeller wear is undoubtedly connected with cavitation are not included in the reliability analyses. If guide vanes are employed, it is assumed they may be repaired at overhaul times.

b. Judgment Factors

The average time between repair or replacement of a worn component is dependent on the plant operator's judgment as to how much wear is tolerable. Tolerance limits are often influenced by demand for pumping time or by wear rates which differ greatly from those expected. In any case, the times between maintenance actions on a particular component should be adjusted to reflect the percentage of wear which motivated them. If the amount of wear at replacement is unknown, a judgment factor will not be used to weight the data, but proper selection of the tolerance should lead to a statistical cancellation of much of the resultant error.

(1) Wear Ring Judgment

The wear rings of a pump will be defined as those rings which restrict the flow of water between the impeller outlet and its suction intake at the eye.

A logical basis for setting tolerance limits on the wear to be allowed in the Tehachapi wear rings would be an economic analysis. This would relate an increase in ring clearance with a loss in overall pumping efficiency. Then, if the rate of wear versus time can be reliably estimated, the efficiency losses may be integrated over time and a cost of power wasted may be obtained as a function of pumping hours. When the cumulative cost of power which can't be utilized reaches a point where it equals the costs of pump time lost, man hours expended and materials needed for ring replacement, this point should mark the useful life of the wearing rings. Actually, these computations should include losses of interstage seal rings as well as wear rings because both types are usually replaced when the pump is dismantled.

A "rule of thumb" standard for wear ring tolerance respected by many pump experts is the one which advises that a set of wear rings should

be replaced when their original clearance has doubled. Examination of field data covering a broad spectrum of pumping demands, wear rates and maintenance-consciousness shows that double the clearance is not a bad average. In addition to efficiency losses caused by wear, many rings are grooved to a depth approximating that of the clearance so that the standard allowed increase should lead to a decrease in relative roughness of the surfaces and a corresponding acceleration of wear rate.

It could be assumed that double clearance in the prototype wear rings represents a loss which should be eliminated for reasons of economy, and field data mean lives could be adjusted to fit times at which wear was double the Tehachapi prototype clearance. However, the prototypes are designed with smaller clearances than are generally found in the plants surveyed so that initial efficiency losses should be lower and proportionate increases will not yield equal monetary losses. In order to use the "rule of thumb" of the pumping industry and to reflect the uncertainties involved therein, it was decided to weight all data on the basis of relative increases in the clearances used at the plants surveyed, rather than absolute increases based on prototype designs.

Although theory indicates that wear rate should have an increasing slope with time, tests on slotted clearances (ref. 7-1) result in an almost linear extrapolation. This may be because the specimens were smooth at the start and roughness which results from erosion increased the friction factor enough to offset velocity increases or the test durations may have been too brief to show a trend. However, in the absence of documented records of wear acceleration, linear extrapolation will be used. Any error introduced in this manner should not be too great if the tolerance is well chosen and clearance at replacement does not fall too far to either side. Twice the original clearance seems to provide a good statistical fit.

(2) Interstage Seal Ring Judgment

The interstage seal rings of a pump will be defined as those which restrict the flow of water between the impeller inlet and the outlet of the previous stage as it exists in the vicinity of the shaft.

As in the case of wear rings, interstage seals should have their tolerance calculated on a cost-effectiveness basis. General practice will be used as the preliminary guide in the belief that experience has supplied a seal wear criteria which will not differ greatly from that which cost analysis might produce.

Interstage seal rings are normally subject to less pressure across their clearance than found in wear rings of the same pump. For this reason, the same percentage of clearance increase in seal rings as wear rings will not produce as great a loss in pump efficiency. Pump manufacturers generally take advantage of this by using less expensive and faster wearing materials in the interstage seals than in the wear rings. When the wear rings are replaced, the interstage seals are also replaced.

Sulzer assumes that the head across the seal ring is about one fourth of stage head while that across the wear ring is about three fourths of stage head (ref. 7-8). If the same efficiency loss is considered the tolerance for wear in either case, the seal ring should withstand three times as much wear as the wear ring. Setting the judgment factor tolerance at six times the original clearance does not fit the data in every case as well as the wear ring judgment factor but it appears to be a good average.

(3) Balance Ring Judgment

The balance rings will be defined as those which restrict the flow of water between the discharge of a single suction pump and its balancing chamber which is normally at suction pressure or close to it.

Data are available which sets the balance ring leakage rates considered tolerance limits for several multistage pumps. These limits average out at about 75% above original leakage which corresponds to a 75% increase in clearance. Most single stage pumps are designed with very little difference between clearance, width, diameter and material of their wear and balancing rings and they are often considered as upper and lower seal rings. However, the wear in the balance ring of a single stage vertical pump tends to proceed at a slower rate than that in the wear ring so that tolerance limits of 75 versus 100 per cent are still usually applicable.

(4) Stuffing Box Judgment

The stuffing box or shaft packing will be defined as that set of seals which restricts the flow of water between suction pressure and atmospheric at either or both ends of the pump.

The criteria by which stuffing boxes are replaced are not dependent on efficiency losses so that tolerances are not readily calculated. The general practice is to replace the box when the leakage comes close to exceeding the disposal capability or on a regular maintenance basis. In two

recorded cases of fixed bushing type stuffing boxes the increases in allowed leakage were 250 and 450%. Because of the wide disparity and lack of further information it is best to assume no judgment factor for stuffing boxes.

(5) Impeller

Impeller erosion is repaired by welding in many plants. It is not possible to set a standard of wear at which time repair should be made. It may be that seasonal pumping plants have more outages for impeller repair than would be necessary for successful operation at a continuous pumping plant but valid data are not available to confirm this. No judgment factor will be used to weight impeller time between repairs. Since this repair is usually done without dismantling the pumps, no significant increase in outage time is foreseen if a judgment factor is not applied.

Impeller replacement time can not be connected with area or depth of erosion because of lack of data and information on the effect of this wear on pump performance. It will be assumed that replacements made due to wear are justified in all cases reported and no judgment factor will be applied.

c. Water Velocity

One mechanism by which the metal parts of a pump erode is the turbulent rush of water past the surfaces of the components. It is found that the rate of erosion increases or decreases as the water velocity relative to the metal surface increases or decreases. This is also true when undissolved solids contribute to the wear since their velocity is dependent on water velocity (ref. 7-2). The components of Tehachapi pumps and all those surveyed in the field are subject to attack by water and solids of varying velocities so that data on component life should be weighted with a water velocity factor.

(1) Rings

Erosion wear of slotted metal samples by clear water is found roughly proportional to pressure across the clearance when the material tested is carbon steel (ref. 7-1). In writing of experience with balancing arrangement design, Karassik and Carter (ref. 7-4) state that one joint designed to operate with total head will present no more problem than several joints distributed throughout the pump because "wear is essentially a function of the pressure drop per inch of running joint length, and if the lengths of these joints are chosen to maintain the same pressure drop per inch, the wear will

not be affected by the number of joints nor by the pressure differential across them".

Stepanoff (ref. 7-3) finds that the pressure or head across a narrow clearance such as that of a pump ring may be expressed as follows:

$$H_L = f \frac{L}{d} \frac{v^2}{2g} + 0.5 \frac{v^2}{2g} + \frac{v^2}{2g} \qquad (7\text{-}10) \qquad (\text{ref. } 7\text{-}3)$$

where H_L = head across the clearance

f = friction coefficient of the metal

L = throttling length

d = diametrical clearance

v = water velocity

g = acceleration due to gravity

The first term of equation (7-10) represents a friction loss, the second an entrance loss, and the third is velocity head at discharge.

Equation (7-10) may be rearranged to get leakage water velocity through the clearance as follows:

$$v_L^2 = \frac{2g\, H_L}{\left(f \frac{L}{d} + 1.5 \right)} \qquad (7\text{-}11)$$

Thus, it is seen that if wear is proportional to head, it should also be proportional to relative water velocity in the clearance taken to the second power. There is some controversy over whether this was true in the tests in ref. 7-1 where two leakage rates were measured, and whether different materials have different reactions. However, it will be shown later that wear in pumps must be much more intimately tied to solid particle erosion than to the corrosion-erosion of interest in these boiler feed tests.

Finnie (ref. 7-2) finds that erosion of metals by particles in a fluid stream is proportional to velocity squared and sometimes to velocity to the fourth power where turbulence is involved. In view of field experience

data outlined later in the report and Finnie's findings, it is assumed that wear rates are generally proportional to water velocity squared until such time as sufficient wear test data using average water proves otherwise.

Equation (7-11) will suffice for simple rings with a straight bore and only one entrance and one exit loss. However, common practice in European pump design is to break the total throttling length (L) into segments arranged as a labyrinth of throttling surfaces of length "l" for each surface. Each throttling surface in the labyrinth has a pressure relief chamber at either end of it so that entrance and exit losses are multiplied by the number of throttling surface segments. Equation (7-11) then becomes:

$$v_L^2 = \frac{2g\,H_L}{z\,(f\,\frac{l}{d} + 1.5)} \qquad (7\text{-}12)$$

where z = the number of throttling surfaces

l = average length of each throttling surface

The above equation agrees almost exactly with results obtained by Sulzer (ref. 7-8) in controlled tests with a variety of labyrinth configurations and pressures. Stepanoff's tests with various ring designs (ref. 7-3) have been given a different interpretation by Denny (ref. 7-9), but European design allows a much larger relative clearance at discontinuities in the throttling surface than were used in these tests.

American ring designs are usually straight so that equation (7-11) applies, but there are instances in which lands with relatively narrow clearances at the discontinuities are used. In these cases, the more refined versions of head loss may be used. That is:

$$h_L = C\,\frac{(v_1 - v_2)^2}{2g} \qquad (7\text{-}13)$$

where h_L = head lost at a sudden enlargement or contraction in clearance

v_1 = velocity of the water before the change in clearance

v_2 = velocity of water after the change

C = 1 for a sudden enlargement or 0.4 to 0.5 for a sudden reduction

For changes in annular clearances as found in rings, it may be assumed that velocity is inversely proportional to clearance. Thus, if the change in clearance is large, equation (7-10) applies, but otherwise it may be modified by (7-13).

A 90° change in the path of the water flow with relatively the same clearance should result in a $v^2/2g$ loss. The effect of two 90° changes or a step in the ring with relatively little change in clearance may be calculated from Stepanoff's equations, including (7-13).

Because the water between the impeller and its shroud is in rotation, the pressure at the wear rings or balance rings is not equal to stage head. It may be calculated from:

$$H_L = H(1 - K_3^2) - 1/4\,\frac{u_i^2 - u_r^2}{2g} \qquad (7\text{-}14) \qquad (\text{ref. } 7\text{-}3)$$

where H = head per stage

K_3 = a design constant for volute area (function of specific speed)

u_i = impeller peripheral velocity

u_r = ring peripheral velocity

Equation (7-14) in most cases results in a head for wear rings of about three-fourths stage head which agrees quite well with empirical measurements. (refs. 7-3, -8, -10). This equation may also be used for balance rings whose outside diameter is not too dissimilar from that of wear rings, but to H figured for the last stage must be added the total head of the other stages.

Pressure decreases slightly between the wear ring and the shaft where the interstage seal ring is located. Head on the impeller discharge side of the seal ring may be calculated from

$$H_{sd} = H_L - 1/8\,\frac{u_r^2 - u_s^2}{2g} \qquad (7\text{-}15) \qquad (\text{ref. } 7\text{-}3)$$

where H_{sd} = pressure at the impeller discharge side of the seal

u_s = peripheral velocity of the interstage seal ring

Head at the impeller suction side of the interstage seal is about the same as head at discharge of the previous stage. The head across the seal ring clearance is

$$H_s = H - H_L + 1/8 \frac{u_r^2 - u_s^2}{2g} \qquad (7\text{-}16)$$

which usually amounts to slightly more than one-fourth of stage head.

Leakage velocity as calculated from the above equations represents only the velocity component perpendicular to the plane of the ring diameter. The water is also given a tangential component of velocity by peripheral motion of the ring. Water in contact with the ring surfaces would have a tendency to either remain stationary or rotate at the same speed as the contacting surface, but narrow clearances, leakage velocity forces, and turbulence would act to hinder the occurrence of relative tangential velocities close to zero. Then too, the erosion by solids whose size is significant compared to the clearance would be a function of some midstream velocity, the tangential component of which might be up to one-half the peripheral velocity of the ring.

The wear tests currently being conducted by DWR and DMJM employ both stationary and rotary specimens so that the difference in wear rate due to tangential velocity will be investigated. In this preliminary report an estimate of the effective tangential velocity will be used and the estimate will be refined as conclusive results become available from the wear test.

The preliminary estimate of effective tangential velocity component is taken from a study of wear in the wearing rings of a pump as compared to wear in the interstage seals of the same pump. Such a comparison removes water quality and number of starts as variables. The pump chosen was one of both average size and water quality as far as the field surveys were concerned. Design parameters, materials, and extent of clearance increase were known. Material weighting factors were taken from the results given in ref. 7-1 and 7-6. For this particular combination of materials, both references agreed on their comparative merits. Leakage velocity squared factors were calculated from the above equations using the known design parameters. Then the relative wear as it should have been on a material basis alone was weighted by a $v_L^2 + x^2$ factor to make it fit actual wear. It

was found that the additional x^2 factor needed was one-eighth of peripheral velocity squared. The total velocity squared term of interest as a weighting factor is then:

$$v^2 = v_L^2 + 1/8 \left(\frac{\pi D_r n}{60}\right)^2 \tag{7-17}$$

where v = resultant water velocity of interest

v_L = leakage component of velocity

D_r = ring diameter

n = rpm of the pump

One all encompassing equation was derived by combining the above equations as follows:

$$v^2 = \frac{64.4 \left[H - H'K + 1.06 \times 10^{-5} n^2 (D_i^2 - D_r^2) + 5.31 \times 10^{-6} n^2 (D'^2_i - D_r^2) \right]}{z \left(\frac{f\ l}{d} + 1.5\right)}$$

$$+ \frac{\left(\frac{\pi D''_r\ n}{60}\right)^2}{8} + \frac{\left(\frac{\pi D'_s\ n}{60}\right)^2}{8} \tag{7-18}$$

where H = head

K = $1 - K_3^2$, a design constant

n = pump speed

D_i = impeller diameter

D_r = ring diameter

D_s = seal diameter

z = number of throttlings

f = friction coefficient

l = length of throttling surface

d = diametrical clearance

Substitution or deletion of the proper parameters and use of the appropriate sign for "z" allows the equation to be used in the various velocity computations for all ring types. The "z" quantity may also be adjusted to fit the rare situations in which bends or narrow pressure relief channels are used and those situations in which design parameters can only be estimated on a comparative basis.

One other device, aside from equation (7-18), was used to calculate water velocity in an effort to make results fit empirical data as closely as possible. Leakage rates for five Sulzer designed balancing labyrinths were known (ref. 7-8). Leakage velocities were calculated directly from these rates and known clearances and diameters. An equation was also given in ref. 7-8 from which relative power losses and leakage rates for a pump could be calculated using the known losses of the five referenced pumps. It is:

$$\frac{P}{P'} = \frac{K}{K'} \left(\frac{n_s}{n'_s}\right)^{3/4} \left(\frac{H_L}{H'_L}\right)^{1/2} \left(\frac{D_o}{D'_o}\right)^{3/2} \left(\frac{Q'}{Q}\right) \left(\frac{e}{e'}\right) \qquad (7\text{-}19)$$

where P = power loss

K = leakage coefficient

n_s = specific speed

H_L = head across the labyrinth

D_o = outside diameter of the labyrinth

Q = pump capacity

e = pump efficiency

primed quantities = referenced pump parameters

The leakage coefficient term may be considered unity when the two pumps being compared are similar in size and it is found that substitution of total head for head across the labyrinth yields almost identical results.

Operational parameters used in all of the above velocity equations such as head, speed, capacity and efficiency were obtained from the survey forms and ref. 7-11. Specific speeds were given or calculated from the same information.

Dimensional parameters such as impeller and ring diameters were obtained for the most part from survey information and drawings. In two cases, impeller diameter was calculated from

$$D_i = \frac{60\,(K g H)^{1/2}}{\pi n} \qquad (7\text{-}20)$$

where K = a constant assumed to be about 0.5.

In several cases wear ring diameter was calculated from

$$D_r = k\,(n_s)^{1/2}\, D_i \qquad (7\text{-}21)$$

where k = 0.017 as an average of ten known cases

n_s = specific speed

A few seal ring diameters were estimated on the basis of ten known cases as being about four-tenths the size of the impeller.

Design parameters such as number (z) and length (l) of throttling surfaces as well as ring clearances (d) were taken from the field surveys and drawings and from correspondence with the model test firms. The number "z" was adjusted in some cases so that 1.5z would represent all losses due to path change and pressure relief, as well as entrance and exit losses.

The coefficient of friction "f" was taken to be 0.04 for European pumps because this is the result which Sulzer's experiments have shown to be empirically correct for Reynolds numbers of interest (ref. 7-8). Comparisons of other European manufacturers' ring designs and achieved

efficiencies for similar pumps indicated that they, too, employed a surface roughness for which the coefficient could be assumed 0.04.

The friction coefficient "f" was assumed to be .02 for all American designed pumps because they employ smooth surfaces for which this is the empirical value at R_e of about 10^5 (ref. 7-3). Reynolds numbers do not vary much for different pump designs because each manufacturer appears to be aiming for a leakage velocity of 50 feet per second or less.

One exception to the friction coefficients outlined above involves the single stage Tehachapi prototype. This pump features grooves on both surfaces of the rings rather than one. The manufacturer declined to state the empirical coefficient of friction at the time of this report. His estimate of balance ring efficiency loss (ref. 7-11), if correct, would indicate extremely high velocity. The v^2 factor was first calculated on the speculation that a 20% increase in friction might be attained with dual grooves. The resultant factor was tempered by averaging it with v^2 factors for other pumps designed in a conventional manner by the same manufacturer.

(2) Impellers

Impellers are subject to erosion by turbulent water and suspended solids much like rings are. However, there has not been much evidence which could directly link rate of impeller erosion with relative water velocity squared. Most tests on impeller wear are concerned only with cavitation erosion. One such test (ref. 7-5) used metal discs with holes drilled through them to create a disturbance which presumably approached that of cavitation. However, the character of turbulence thus created is probably not unlike that which exists when the leading edge of the impeller blade strikes the water in the suction. Relative water-disc velocities were noted along with volume of material lost in a given time. When these results are plotted for the same material it appears that v^2 quite appropriately suggests wear rate at a relative speed of 125 feet per second. This relative speed is found to approximate that of the Tehachapi prototype impellers and makes a fair average for other pumps surveyed.

It may be assumed that prerotation of the water in the suction is not a significant factor by design (ref. 7-7). Then the water velocity relative to the impeller inlet becomes

$$v_i = \left(c_{m1}^{2} + u_1^{2} \right)^{1/2}$$

and $$v_i^2 = c_{m1}^2 + u_1^2 \qquad (7\text{-}22) \quad (\text{ref. } 7\text{-}3)$$

where v_i = relative velocity of flow at the impeller inlet

c_{m1} = absolute velocity normal to the peripheral velocity

u_1 = peripheral velocity of the impeller eye.

The absolute velocity "c_{m1}" is known as the "throughput" and is found from

$$c_{m1} = \frac{Q}{\pi D_{1m} d_1} \qquad (7\text{-}23)$$

where Q = capacity

D_{1m} = mid diameter of impeller leading edge

d_1 = average diameter of the cross sectional flow about the mid point

The dimensions "D_{1m}" and "d_1" may be measured from drawings of the pump.

The peripheral velocity of the impeller eye is simply

$$u_1 = \frac{\pi D_1 n}{60} \qquad (7\text{-}24)$$

where n = rpm

D_1 = impeller eye diameter

Where D_1 or c_{m1} could not be ascertained from surveys or drawings it was obtained by means of plots for ten known points rather evenly distributed over a range of specific speeds from 1000 to 2300 and from known impeller diameter "D_2" and head. It turned out that D_1/D_2 was slightly higher at all speeds than claimed by Stepanoff (ref. 7-3). However, his plot of K_{m1}, from which c_{m1} is found $\left[c_{m1} = K_{m1} (2gH)^{1/2}\right]$, proved to fit real data quite well.

In case it is not apparent why the inlet velocity triangle is of interest rather than the discharge triangle, it is because wear is usually most severe at the inlet due to high local velocities, impact pressures and turbulence.

(3) Stuffing Boxes

The stuffing boxes covered in the survey are of four general types. The familiar adjustable soft pack is used in slow speed American pumps. Common practice in Europe is to use fixed bushing types of seals with either carbon rings or babbit bushings. There is one case in which a mechanical carbon seal is used.

Great variety in stuffing box design and the large number of design parameters for each resulted in a situation in which none of the surveys and drawings gave enough reliable information on which to base a v^2 weight factor. For soft packing, degree of tightening could not be determined. In fixed bushings, seal water pressure was unknown and clearance of either the labyrinth or shaft seal, or both, were missing. There was also no way to account for rubbing wear which quite possibly occurs with the narrow clearances used. For these reasons and others, a v^2 factor was not calculated for stuffing box field data.

The Tehachapi prototypes feature fixed bushings and a mechanical seal. No mean life data are available on the mechanical seal used in the two-stage pump or on the straight bore, double-grooved design used in the single-stage prototype. Each prototype was, therefore, assigned a conventional fixed bushing design similar to that found in the field surveys and the four-stage prototype. Mean life from the surveys was applied to the four-stage prototype box. Then the same ratio between stuffing box life and wear ring life for the four-stage pump was assumed to hold for the one and two-stage pumps. Increased submergence for these prototypes indicated that v^2 might also increase as it did in wear ring calculations.

d. Materials

In the interim period before conclusive results are obtained from the DWR-DMJM Wear Test Program, it is necessary to accept the work which has already been done in the field of material wear. The two wear tests which seem most applicable to the pump erosion situation are those detailed in ref. 7-1 and 7-6. However, both of these tests leave much to be desired.

The corrosion-erosion tests of boiler feed pumps (ref. 7-1) were conducted using very hot, neutral water with probably no significant solids content. On the other extreme, the study of material erosion by sand (ref. 7-6) used a solution which was two parts sand and one part water. Neither of these tests represents the conditions found in typical water pumping as found in the field or expected at Tehachapi.

The chief mechanism for wear in the boiler feed test was simply corrosion aided by turbulent water flow. Under these conditions, it was found that SAE 1020 steel lost about 250 times as much material weight as 12% chrome steel in the same amount of time. The sand erosion test yielded only 1.4 times the loss for the same materials comparison. Bronzes showed pretty much the same relative resistance in either test. Table 7-I gives the relative erosion weight loss results using 12-13% chrome steel as the standard.

A simple device was used to determine whether the chief wear mechanism encountered in pumping plants was water or solids. Slotted area attacked in the water corrosion-erosion tests was known. From known densities of materials and known original slot clearance, weight loss needed to double a 0.025 inch clearance was calculated for materials of interest. Then surveyed component lives, as weighted by judgment and velocity factors, were used in conjunction with test results on weight loss versus time to determine what percentage of the needed weight loss might be attributed to water corrosion-erosion alone. In almost all cases, it was found that water alone was an insignificant factor as a wear mechanism in pumps. In isolated cases of cast steel rings, a few percentage points of the wear might be attributed to corrosion providing a pH of about 8.4 is assumed and relative loss weighted by results in ref. 13 on loss versus pH. In any case, the percentages found were less than the normal variance in weight loss from repeated tests. Therefore, it was decided to use relative sand erosion figures which seemed to fit the data quite well.

Materials chosen for the wear rings and balance rings in the prototypes were 1020 and hardened 1040 steels. It must be emphasized that these materials were not chosen because they represent the optimum selection. Such a selection must await results of the wear test and further analyses of other test and field data. They were chosen because they are popular in contemporary American pump and turbine plants, they are not expensive, they show good sand erosion resistance, and they have been tested in actual service.

MATERIALS	RELATIVE LOSS WATER CORROSION-EROSION	RELATIVE LOSS SAND EROSION
12-13% Cr	1	1
1020 Steel	250	1.4
1040 Steel BH 440		0.95
1040 Steel BH 170		1.25
Cast Steel Casing	167	1.4
Cast Iron (Avg)	34	2.66
Al. Bronze (Ampco 18)		2.1
Mn. Bronze		2.8
Ni. Bronze	3.1	
Si. Bronze	14	
Navy M	2.1	
Bronze (Avg)		2.5
Yellow Brass		3.4
.8 Cr .7 Ni .3 Mo Steel	4.1	

TABLE 7-I RELATIVE SAND AND WATER EROSION

Since the water at Tracy Pumping Plant proved to have an average pH of about seven, it was assumed that no significant portion of the ring wear at Tehachapi would be attributable to corrosion.

Thirteen percent chrome steel was assumed for the impeller since the majority of field surveys listed this material, but the popular babbit and cast steel interstage ring materials were passed over in favor of bronze and cast iron. The reason for not choosing the softer interstage materials was that data indicated their mean life might not exceed that of the wear ring lives weighted for 1020 and 1040 steel materials, and it is desirable to change wear and seal rings simultaneously. Babbit has not been subjected to controlled wear tests so that data are not available on its relative resistance to sand erosion. However, two plants surveyed used bronze and cast iron in their wear rings and a babbit-cast steel combination for their interstage seal rings. Mean lives for one plant could be weighted for both velocity and judgment, while the other could be weighted for velocity only so that relative lives were weighted by two and one, respectively, to arrive at an average relative life of 3.5 for the bronze-cast iron over the cast steel-babbit. The consequences of this factor being in error are not great since overhaul time for the seal rings is to be determined from life of the wear rings. Sufficient proof that bronze-cast iron seal rings will outlast steel wear rings is found in the data.

Stuffing box life was not weighted for material for the same reason it was not weighted for judgment and velocity; a lack of detailed data. Most of the fixed bushing types have a chrome and bronze labyrinth followed by a babbit-steel or babbit-chrome bushing for one end of the pump and the bushing alone for the other end.

e. Special Factors

Special weighting factors which deserve consideration are continuity of operation; ring eccentricity; settling, flushing, and centrifuging of solids; and water quality. Analysis shows that the last of these factors is so dominant, indefinable, and unpredictable that the others cannot and need not be determined for data weighting purposes.

It is a common assumption that a pump which is started and stopped quite frequently in the course of its operation should experience more trouble than one which suffers few interruptions to its operation. For this reason, the field surveys included questions on the number of starts per month or day, length of the pumping season, hours pumped per day and year, and so forth, so that estimates could be made of the number of starts per

thousand operating hours. These estimates range from two to 1500 with the average being about 200. The plants surveyed were arranged in three groups according to water quality with component lives associated with them. These lives were weighted for judgment, velocity, and material, and included times the component had operated without failure. No correlation could be found between component life and number of starts within the groups representing fairly uniform water quality.

Excellent records kept at Tracy Pumping Plant made it possible to study wear versus ring eccentricity. The records showed north, south, east, and west measurements of ring clearances at installation and at removal for overhaul. Accumulated operating hours and unit starts could also be determined at any time. The only variable in the wear of twelve rings was eccentricity. Comparisons were made between the wear per 30,000 hours and the amount the rings were initially out of round, which ranged from .002 to .026 on an average clearance of .035. No correlation could be found.

While a pump is idle, suspended solids in water in the volute and cross-overs settles out and accumulates at the lowest points on the discharge side of the pump which are in the vicinity of the wear rings for a vertical pump (ref. 7-4). If this phenomenon were significant, it could lead to longer life for a horizontal pump's wear rings than for those of a vertical pump operating under similar conditions. No such trend is definitely indicated by the data, nor is there any obvious connection between use of seal water flushing and greater ring life.

Other observations of multistage pumps, whether vertical or horizontal, indicate the rings nearest the suction inlets wear faster than rings in later stages if the water contains a fair concentration of solids. It is thought that solids in the inlet are fairly uniformly distributed while the centrifugal action of each stage progressively forces the particles further out toward the periphery of the volute or cross-over and away from the rings. In this case, the multistage pump might be favored because losses in overall efficiency due to leakage may not increase as rapidly as they would for single stage pumps in the same application. Evidence of this is also impossible to filter out of the amount of data at hand, some of which are contradictory in nature.

The subject of water quality and its effects on component life is so complex as to make scientific description and classification difficult, if not impossible. Variables to consider are parts per million of undissolved solids (to which wear should be proportional) (ref. 7-2), hardness and size of the

suspended solids, pH, corrosion index, salt and CO_2 content, specific conductance, and the component materials which are in contact with the water. Field evidence indicates that concentration of solids is the variable most closely and dominantly related to wear.

Survey descriptions and laboratory analyses of water quality at the various plants were studied in an effort to categorize the water. An attempt was made to ignore mean life of the components at each plant when drawing up the categories, but it is probable that a plant operator's description of water quality is influenced by his maintenance experiences. Even among impartial observers, opinions on what is clear water and what is dirty water vary considerably. For this reason, fine distinctions in water quality were not attempted and only three categories were formulated.

Good water is considered to be that in which visible solid content is not particularly notable. This category includes all the American plants visited except Tracy, as well as plants with "exceptionally clear" water such as Etzelwerk, Provvidenza, and Sipplingen, and "good" water as found at the Hausern complex.

The average water classification includes all those plants with some solids in their water, but not notable amounts. Most of the glacial silt waters are in this category whether high or low in altitude and regardless of the relative compositions of limestone versus quartz. Tracy water, because of its turbidity, is considered to be average.

The poor quality water category encompasses only six plants. Typical of the plants and comments describing the associated water is Motec and "poor, sharp glacial silt, much sand in year 1963". Other plants included are Arolla, Stafel, and Ferpecle.

Mean lives and times without failure for wear rings, balance rings, and impellers at the various plants were listed in their respective quality groups. Only three notable exceptions to the failure time-water quality correlations were found. One of these cases, Lewiston wear rings, was eliminated from the mean life-water quality comparison. This was because, while the wear rings eroded excessively in a short time, the upper or balance rings exhibited no measurable increase in clearance. Water quality was thus eliminated as a factor in this case.

Average mean times to failure were calculated for rings and impellers in each water quality group. It was found that impeller mean life

for average water was about half the average for good water and four times that for poor water. The balance and wear rings exhibited a proportionate but greater separation than the impellers. In order to avoid compounding of errors by over-correction, the weight factors one-half and four were adapted.

It was decided that the water at Tehachapi would in all probability fit into the average quality bracket since the forebay is not large enough to afford the plant more than one-half hour of capacity pumping and since Delta water is quite generally colored and clouded with suspended sediment. It was also assumed that the water would be neutral as it is in the region of the Delta near Tracy.

2. Computation of Weighted Component Lives

An example will be given of the computation and application of weight factors. The sample data chosen for this demonstration of weighting are from Motec and are neither greatly detailed and accurate nor unusually obscure. Motec is chosen because it contains all component types which were weighted and has failure figures for most.

No clearances were given on the seal and wear rings at replacement but leakage for the balance labyrinth at start and replacement was given as 30 liters per second and 55 liters per second. This was close enough to a 75% increase so that failure times were not adjusted in this case except that a needed repair was not made in the Winter of 1963 for lack of spare parts but the time elapsed since the previous replacement was taken as a time between failures. The times to failure for the balance rings were 2300, 3000, and 1500; for the wear rings: 5000 and 2000; for the seal rings: 5000 and 2000. Times without failure were 7900 for the fixed bushing stuffing box, 1000 for the wear rings and seal rings, and 7900 for the impeller. Times in all cases were rounded off to the nearest hundred.

Velocity squared for the Motec rings was calculated by hand and also as a part of the computer program for a double check. Equations (7-12), (7-14), (7-16), (7-17) and (7-18) were used in these computations with the following parameters:

H = head per stage = 688 ft.

H_t = pump head = 2065 ft.

K = design constant = 0.84

D_i = impeller diameter = 5.2 ft.

D_r = wear ring diameter - 3.5 ft.

D_b = balance ring outer diameter = 3.34 ft.

D_s = seal ring diameter = 1.66 ft.

Z_r = number of wear ring throttlings = 4

f = friction coefficient of rings = 0.04

l_r = average length of wear ring throttling = 0.14 ft.

d_r = diametrical wear ring clearance = 0.00492 ft.

n = pump speed = 750 rpm

l_s = average length of seal ring throttling = 0.75 ft.

Z_s = number of seal ring throttlings = 1.0

d_s = diametrical seal ring clearance = 0.00394 ft.

D_{bm} = balance ring mid diameter = 2.6 ft.

Z_b = number of balance ring throttlings = 11.0

l_b = average length of balance ring throttling = 0.29 ft.

d_b = diametrical balance ring clearance = 0.00492 ft.

Results obtained on the computer and by hand respectively are as follows (in hours):

wear ring = 5348.7235, 5360

seal ring = 2134.5351, 2150

balance ring = 3791.4699, 3920

The two sets of results are within slide rule and round-off accuracy for the first two outputs. The balance ring computations were done in two different manners. The computer figure came from the general ring equation, (7-18), while the hand calculation was simply figured on the basis of given leakage and measured dimensions. The variance is only about three percent which is considered to be a very good confirmation of weighting techniques used, particularly when it is noted that real operational clearance is generally on the high side of design clearance and the leakage figure was rounded off.

Velocity squared for the impeller was calculated by hand from equations (7-22), (7-23) and (7-24) and turned out to be 12, 765. Impeller v^2 for the four stage Tehachapi prototype was calculated to be 17, 830 so the weighting factor for this impeller becomes 12765/17830 or 0.715 and the 7900 hours without failure at Motec becomes (.715) (7900) or 5700 hours without failure for the four stage prototype at Tehachapi.

Ring v^2 for the Tehachapi four stage prototype was computed to be 4150 for the wear rings, 1395 for the seal rings and 6720 for the balance rings. The first two were found on the computer and the last done by hand using design dimensions and the manufacturer's estimated power loss converted into leakage velocity for the balance labyrinth. Weight factors for Motec data on the basis of this prototype are 5360/4150 for the wear rings, 2150/1395 for the seal rings and 3920/6720 for the balance rings resulting in the following weighted times in thousands of hours:

wear rings: 6.5 and 2.6 to failure, 1.3 without failure

seal rings: 3.1 and 7.7 to failure, 1.5 without

balance rings: 1.4, 1.8, and 0.9 to failure

Material weight factors were applied to the wear rings and balance rings but the impeller and seal rings proved to have the same materials as those presumed for the Tehachapi prototypes. The combination of 1020 and 1040 steel gives an increased life factor over bronze and cast iron of (2.66 + 2.5) / (1.4 + 0.95) or 2.4 (see Table 7-I). The last two failure times for the balance rings listed above were obtained with 13% chrome substituted for the cast iron. The weight factor for this combination becomes (1 + 2.5) / (1.4 + 0.95) or 1.49. Mean lives become:

wear rings: 14.3, 5.7 to failure, 2.9 without

seal rings: 3.1, 7.7 to failure, 1.5 without

balance rings: 3.1, 2.7, 1.3 to failure

impeller: 5.7 without failure

When the data are analyzed on a general water quality basis it is found that Motec belongs to the group with poor water in which average mean lives are less than one-fourth those in the average water group. This last factor, when applied to the above weighted data yields:

wear rings: 57.2 and 22.8 to failure, 11.6 without

seal rings: 12.4, 30.8 to failure, 6 without

balance rings: 12.4, 10.8, 5.2 to failure

impeller: 22.8 without failure

3. Computation of Component Life and Repair Time

Presumably, if the weighting factor derivation were complete and correct, data on failures of all types of rings could be weighted and used in calculating mean life of any one type. However, when wear in the wear rings and balance rings of a single stage pump are compared it is found that velocity factors alone do not account for the differences and in many cases this should be the only variable. Evidently, the situation or duty of a ring in the pump has something to do with wear rate so only wear ring data are used to compute wear ring life and only balance ring data to compute balance ring life, and so forth.

The example given here is that of wear ring life computation because it is applicable to all three lift concepts. A total of 172 times to failure or times without failure for wear rings were extracted from the data. When failure time was given as an average time for some number of pumps in the plant this time was considered the same number of times as the number of pumps to which it applied. That is, if failure time was the average for a plant with six pumps, it was considered that six failures took place at this average time. However, time between ring replacements was assumed to be dictated by the ring which erodes most readily so only one failure per pump per replacement time was allowed. Times were all weighted for the four stage Tehachapi prototype.

Table 7-II lists the times at which failures took place (t_i), the number of operating times greater than t_i (N_i), the number failing at t_i (f_i), the estimated reliability at t_i ($R(t_i)$), the normal deviate of the reliability estimate (y_i), and needed sums and products.

The shortest time to failure in the weighted wear ring data was 13.2 thousand hours. There were 55 times without failure shorter than 13,200 hours so that the number of observations left at this time was 172 minus 55 or 117. Two failures occurred at this time so that the reliability estimate was (117 - 2 + 1) / (117 + 1) as per the first term of equation (7-1). The "y_i" was found in a table of area under the normal density function for a reliability of .984. It is negative because it occurs before the 0.5 reliability point on the curve.

The second shortest failure time was 14,900 hours but only 104 survivors were left at this time because of the two which failed at 13,200 and eleven more whose incomplete operating times lay between 13,200 and 14,900. Survival rate in this interval was (104 - 6 + 1) / (104 + 1) or .945 and this multiplied by the reliability at 13,200 gave a reliability of .928 as per the first two terms of equation (7-1). Again the deviate, -1.46, was found in the normal area table for a reliability of .928.

At 17,200, there were 89 times left, one of which was a failure time. Then $R(t_i)$ is (89 - 1 + 1) (.928) / (89 + 1) for 17,200 hours. This process continues in order of shortest times first until the longest time to failure is accounted for.

Equations (7-3) and (7-4) are used to get a mean life (m) and standard deviation (s) which are 31,500 and 16,400 respectively for the Tehachapi four stage prototype. The 95% confidence limits are 27,700 to 35,300 as calculated from equation (7-2) where f is the total number of failures entering into the data or 70.

Wear ring lives for the other prototypes were calculated using velocity squared weighting factors. The average velocity squared for the single stage pumps as calculated in accordance with the procedure outlined previously under "Water Velocity", is 4510. The four stage wear ring v^2 was 4150. Therefore, wear ring life in the three lift concept is about (4150) (31.5) / 4510 or 29,000 hours. For the two lift concept it is (4150) (31.5) / 4200 or 31,200 hours.

t_i hours	N_i	f_i	$R(t_i)$	y_i	y_i^2	$t_i y_i$	$f_i t_i$
13,200	117	2	.984	-2.15	4.62	-28.4	26.4
14,900	104	6	.928	-1.46	2.14	-21.8	89.4
17,200	89	1	.918	-1.39	1.93	-23.9	17.2
18,600	87	1	.908	-1.33	1.77	-24.8	18.6
19,800	84	1	.897	-1.26	1.59	-25.0	19.8
20,400	80	16	.720	-0.58	0.34	-11.8	326.4
20,800	64	5	.665	-0.43	0.19	- 8.9	104.0
21,500	59	2	.643	-0.37	0.14	- 8.0	43.0
21,900	51	1	.631	-0.34	0.12	- 7.4	21.9
22,400	50	1	.620	-0.31	0.10	- 7.0	22.4
22,800	49	1	.609	-0.28	0.08	- 6.4	22.8
24,500	48	1	.596	-0.24	0.06	- 5.9	24.5
27,600	39	1	.582	-0.21	0.04	- 5.8	27.6
28,000	38	6	.493	0.02	0.00	0.6	168.0
30,900	32	3	.448	0.13	0.02	4.0	92.7
35,200	27	4	.371	0.33	0.11	11.6	140.8
36,000	23	2	.340	0.41	0.17	14.7	72.0
44,400	17	6	.227	0.75	0.56	33.2	266.4
51,000	11	4	.151	1.03	1.06	52.5	204.0
57,200	7	1	.133	1.11	1.23	63.5	57.2
78,500	6	4	.057	1.58	2.50	124.0	314.0
78,700	2	1	.038	1.77	3.14	139.5	78.7
705, 500		70		-3.22	21.91	443.6	2157.8

TABLE 7-II RELIABILITY ESTIMATE DATA

Repair times were quite variable. The following are numbers of repairs made and time for repairs in hours (assuming three shift maintenance):

. multistage wear and seal ring replacement	3 at 380 hours 10 at 120 hours
. multistage seal and balance rings	1 at 253 hours
. multistage wear and seal ring plus impeller welding	2 at 336 hours 1 at 120 hours

The above times were averaged to get a multistage wear and seal ring replacement time of 200 hours.

The following times referred to impellers only:

. multistage impeller repair	19 at 40 hours
. multistage impeller replacement	19 at 120 hours

A common procedure in pumping plants is to repair the impeller at two overhauls and replace it every third overhaul. If this practice were carried out it would mean an average impeller maintenance time of (120 + 80) / 3 or 67 hours.

This time could be added to the ring replacement average of 200 hours to make a total overhaul time but it should be noted that an impeller repair and balance ring repair already figure in to the ring average and that assembly and dismantling time are duplicated in the ring and impeller change times. Therefore, an overhaul time of 250 hours for multistage pumps is considered to be on the conservative side.

The single stage pump repair times were as follows:

. ring replacement	6 at 370 hours 7 at 194 hours
. ring replacement plus impeller repair	3 at 40 hours 20 at 136 hours

The average here was 178 hours for ring replacement. Typical of other impeller repair times was 70 hours. The overhaul time for single stage pumps was set conservatively at 230 hours.

Times for fixed bushing packing replacement were two at 60 hours, two at 48 hours and two at 64 hours for an average of 47 hours which was rounded off to 50 hours for the study.

Balance ring repair does not require that the pump be completely dismantled, and complete dismantling and ring replacement including balance rings was listed at 253 hours on a pump having three wear rings and two seal rings. It was assumed that balance ring repair time alone would be about 100 hours.

4. Preparation of Planned Outage Schedule

Mean time between replacements of wear rings was used as the nucleus of all planned outage schedules. For example, it was noted that if the balance ring life in the four stage pump were adjusted from 20,400 to 21,000 hours, every third balance ring repair would coincide with every second ring overhaul and maintenance time could be conserved. The stuffing box life was boosted from 13,700 to 14,000 hours so that every ninth packing repair would coincide with a fourth overhaul and every third packing repair would occur simultaneously with the second balance ring repair. It was assumed that impeller repair could be delayed to overhaul time. The following planned outage schedule for the single lift concept emerged:

Pump Time-hrs.	Repairs	Repair Time-hrs.
14,000	Stuffing Box	50
21,000	Balance Ring	100
28,000	Stuffing Box	50
31,500	Seal Rings, Wear Rings, Impeller	250
42,000	Stuffing Box and Balance Ring	100
56,000	Stuffing Box	50
63,000	Balance, Seal and Wear Ring, Impeller	250
70,000	Stuffing Box	50
84,000	Stuffing Box and Balance Ring	100
94,500	Seals, Wear Rings, Impeller	250
98,000	Stuffing Box	50
105,000	Balance Ring	100
112,000	Stuffing Box	50
126,000	Seals, Wear Rings, Balance Ring, Impeller, Stuffing Box	250

For the one and two stage pumps the wear ring time was taken as overhaul time for all rings and impellers and stuffing box replacement time was again adjusted so that every ninth replacement would coincide with every fourth overhaul. These times are tabulated in Section 7E, Table 7-V.

5. Unplanned Outage Analysis

Percentage of probable unplanned outage time per plant was given in ref. 7-11 and these results have been checked by independent calculations. However, in order to calculate reliability, it was necessary to know the mean time between unplanned outages. The only reasonably detailed and comprehensive data on hand which could be used to determine frequency of these outages were those supplied by the Metropolitan Water District on maintenance for its five aqueduct plants.

All outages except planned overhaul and packing maintenance were counted for each plant through fiscal year 1963. Packing maintenance, though not planned at these plants, is expected to be planned at Tehachapi where fixed bushing types will be used rather than adjustable soft pack. The total pumping hours for each plant through fiscal year 1963 are obtained from annual reports.

The unscheduled outages are (not including packing problems):

Plant	Motor	Motor Bearing	Pump	Pump Bearing
Intake	9	3	2	5
Gene	7	2	2	6
Iron Mountain	0	1	0	2
Eagle Mountain	8	1	0	5
Hayfield	3	11	0	6
Total	27	18	4	23

The pumps at MWD experienced several outages for cracked seal rings. For the purposes of this report, it is assumed that only four (4) of these pump failures could be expected at Tehachapi since the remainder of the outages were due to the experimental nature of the replacement rings.

These outages (given above) were analyzed from the standpoint of double flow, two stage and single flow, four stage to determine the relative number of outages expected from these more complex pumps. It was assumed that the basic design of the motor, bearings and other individual components would be equal in quality for the three types of pumps. The failures attributed to pumps of the single stage type were then weighted to account for the greater number of bearings, seal rings, seals and balancing rings in the two and four stage pumps. Total outages for the three types of pumps for purposes of the Tehachapi analysis are:

	Motor	Motor Bearing	Pump	Pump Bearing	Balance Ring	Total Failures
1-lift (4 stage)	27	18	8	48	4	107
2-lift (2 stage)	27	18	8	48		101
3-lift (1 stage)	27	18	4	23		72

The above data resulted in the following mean times between failures:

	Failures	Mean Time Between Failures (hours)
1-lift	107	21, 100
2-lift	101	22, 300
3-lift	72	31, 000

It was found that the mean time for correcting unplanned outages was .31% for the single stage pumps at Tehachapi,

$$(.31\%) \times (31,000)\ \frac{8760}{8350} = 101.8 \text{ hours}$$

NOTE: 8760 hours in a year
8350 operating hours in a year for Tehachapi.

The time to repair a single stage pump was weighted according to complexity and the added time to repair the balance ring, added bearings and seal rings. The following repair times for unscheduled outages resulted:

	Mean Time Between Failure	Repair Time For Unscheduled Outages
1-lift	21,100	105.3
2-lift	22,300	99.6
3-lift	31,000	101.8

A check was made to see that reliability considering only unplanned outages followed an exponential curve as assumed in predictions made. Data points for the Intake Pumping Plant are shown in Figure 7-4 together with an exponential drawn using the arithmetic mean of 20,000 hours found for the plant. A better fit might have been achieved had the mean time been adjusted to 18 or 19,000 hours, but the points at least show the exponential trend.

Another check was made on mean time between unscheduled outages and mean repair time by analyzing data covering nine years of operation at Grand Coulee Pumping Plant. This plant has 65,000 HP motors which should be more representative of Tehachapi than the MWD motors. Again unscheduled outages for packing problems were ignored. Mean time between outages was found to be 4350 hours. Unscheduled repair time was given for each outage and the mean of these values was about 80 hours. Since the pumping hours involved in this analysis amounted to only about 100,000 while the data from the MWD plants was based on about 2,300,000 operating hours, it was realized that the mean computed before was not significantly reduced if the Grand Coulee data were averaged in.

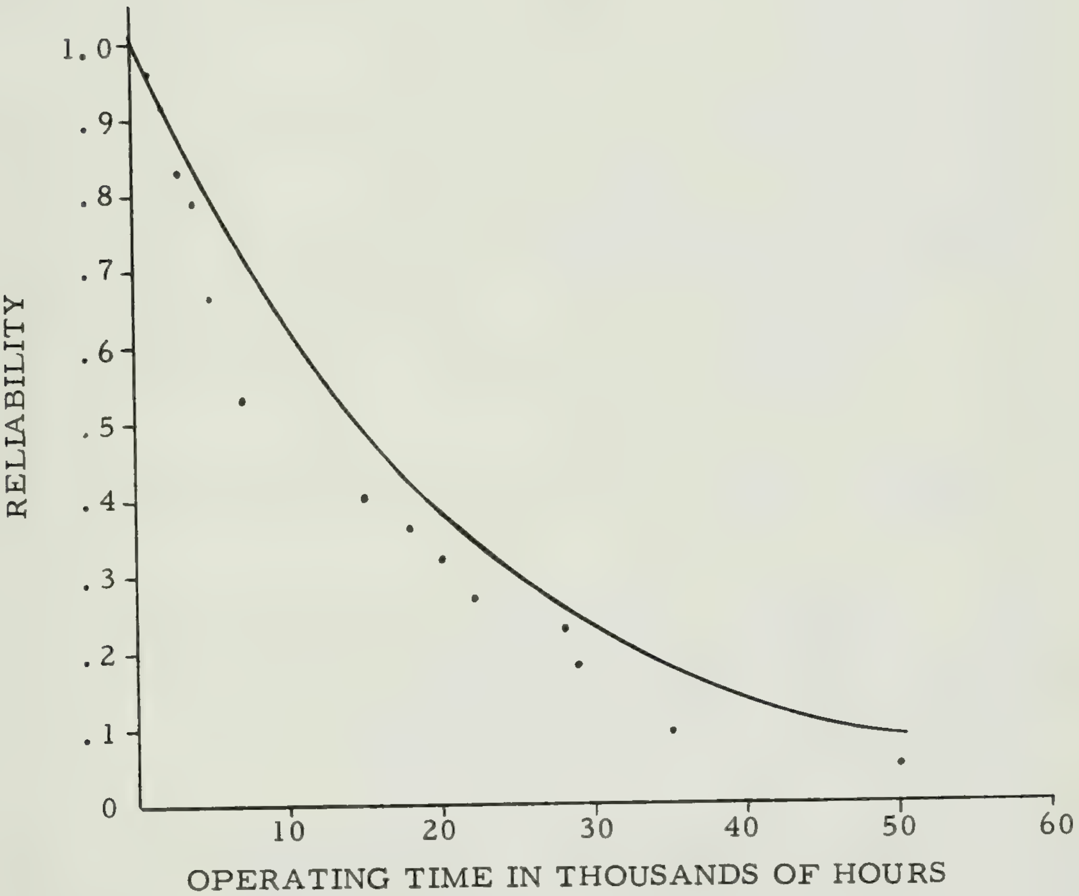


FIGURE 7-4 INTAKE PLANT RELIABILITY FOR UNSCHEDULED OUTAGES

6. <u>Computation of Lift Concept Effectiveness</u>

The equation used in computing lift concept effectiveness is (7-9). The terms of this equation are found from equations (7-8) and (7-5).

Again taking the one lift concept as an example, operating hours are 8350 and hours devoted to scheduled overhaul are (8350)(1700)/126, 000 or 113 hours. There are 297 hours left in the year for unscheduled repairs if it is assumed that Tehachapi operation is independent of unscheduled outages in upstream plants. The mean unscheduled outage time is 105 hours so 297/105 or 2.83 repairs may be made annually. Concept mission reliability is then exponential (-8350/21, 100) multiplied by 1.484 as found for 2.83 repairs on Figure 7-5. The resulting one lift reliability is (0.672)(1.484) or 0.997.

Operational readiness or availability of the one lift concept is 8760 minus 113 hours scheduled outage minus 41 hours unscheduled outage divided by 8760 or 0.982.

One lift concept effectiveness is (0.997)(0.982) or 0.980 for 5000 cfs plant capacity. Increases in plant capacity reduce the number of pump hours per year which increases available repair time annually and therefore increases both concept reliability and operational readiness. Values of effectiveness for various overcapacities are given in the tabulation of results.

The three lift concept puts three plants in series rather than one as with the one lift concept. Annual repair time allowance is (8350)(1320)/ 116, 000 or 95 hours, leaving 315 hours per year to complete unscheduled repairs. This allows 315/3 or 105 hours repair in each plant of the three lift since an outage in one forces equal outage in the others. Reliability of each plant in the lift is (0.766)(1.27) or 0.973 but since all three must operate in series the lift concept reliability is $(0.973)^3$ or 0.921.

Availability of the three lift is 8760 - 95 - (3)(.0031)(8760) divided by 8760 or 0.979 so that lift concept effectiveness is 0.902.

Two lift concept effectiveness is computed in a similar manner.

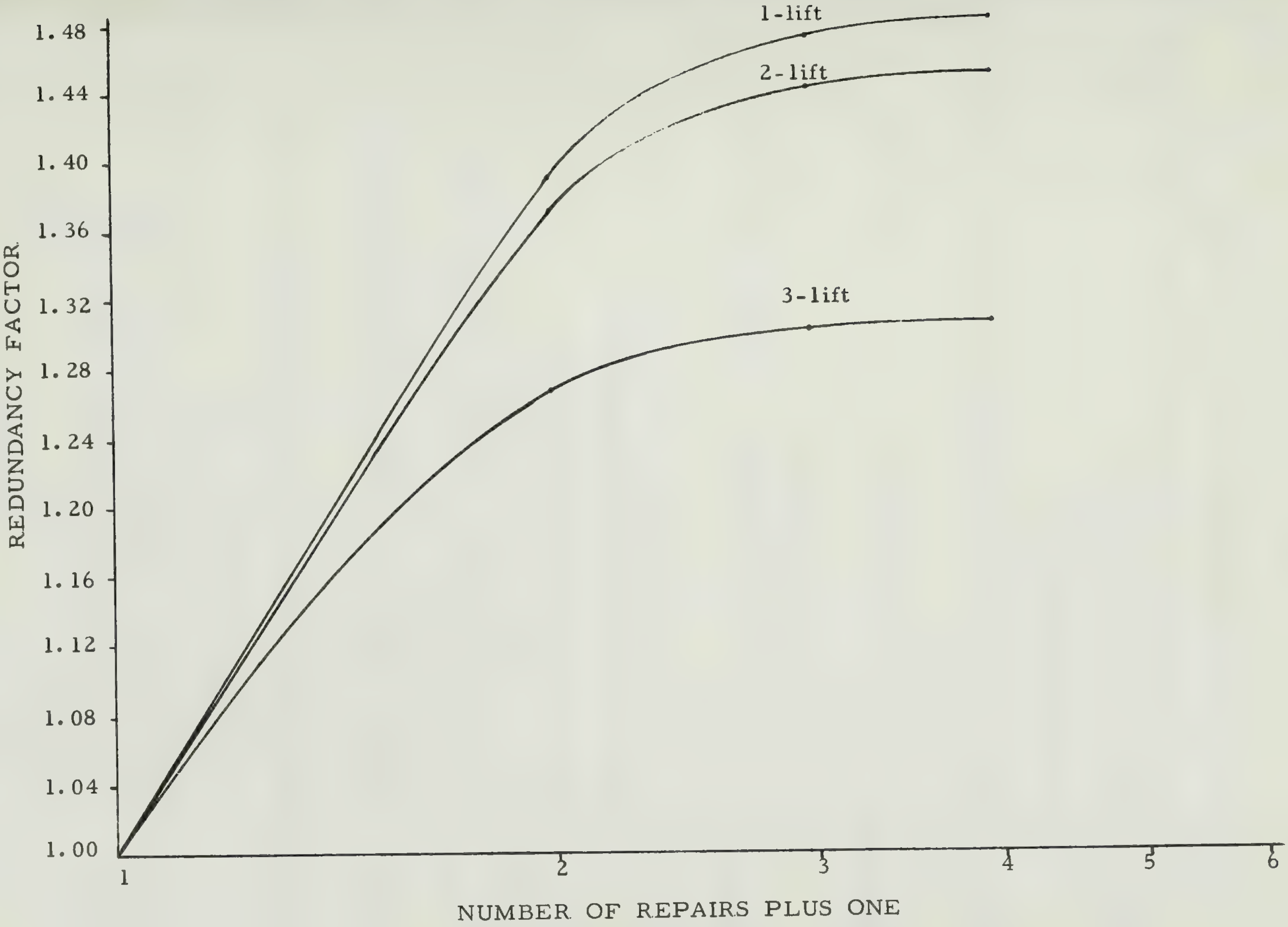


FIGURE 7-5 REDUNDANCY FACTOR FOR REPAIRS

F. RESULT TABULATION AND QUALITATIVE ANALYSIS

This section of the report presents tables of results of the reliability study, an analysis to test their validity, and guidelines for making a choice of lift concepts and pump designs.

1. Component Mean Lives

The following tables present predicted component mean life (time between repairs or replacements) for Tehachapi Prototypes on the basis of the following assumptions.

1. Water quality is average, pH about 6-7, about the same undissolved solids content as Tracy water.

2. Cavitation does not take place.

3. Wear and balance ring materials are SAE 1020 and 1040 hard steel.

4. Interstage seal ring materials are bronze and cast iron.

5. Suction impellers are 13% chrome steel.

6. Wear rings are out of tolerance at twice the original clearance of the average wear ring surveyed in the field.

7. Balance rings are out of tolerance at 1.75 times the original clearance of the average balance ring surveyed in the field.

8. Packing is of the fixed bushing type with babbitt bushing, cast steel sleeve and a four throttling surface labyrinth of 13% chrome and bronze.

9. Interstage seal rings are out of tolerance at six times the original clearance of the average seal ring surveyed in the field.

10. Water velocity squared is as noted in the table.

Mean lives and confidence limits are calculated for components of the four stage pump. These mean times are adjusted by water velocity squared for the two-stage and single-stage pumps. The same velocity factors are used for determining packing life as for wear rings.

LIFT CONCEPT	COMPONENTS	(WATER VELOCITY)^2, Feet^2 / Sec^2	MEAN LIFE Hours
One Lift (4-stage, single-flow, vertical pumps)	wear rings	4150	31,500 ± 3,800
	interstage seals	1395	67,800 ± 18,900
	balance rings	6720	20,400 ± 2,500
	suction impeller	17830	28,500 ± 4,500
	fixed packing		13,700 ± 1,200
Two Lift (2-stage, double-flow, vertical pumps)	wear rings	4200	31,200
	interstage seals	1510	62,500
	suction impellers	15255	33,400
	fixed packing		13,500
Three Lift (1-stage, single-flow, vertical pumps)	wear ring	4510	29,000
	balance ring	4510	30,400
	impeller	14580	34,900
	fixed packing		12,600

TABLE 7-III PREDICTED COMPONENT MEAN LIVES

2. Validity of Component Predictions

Validity of the component mean life predictions depends on validity of the data, the weighting factors, and assumed Tehachapi conditions.

Although every effort was made to base the data on written records and official drawings, this was not always possible, particularly for data from Europe, and a margin of accuracy exists because of this. A measure of data validity is obtained by comparing data points with theoretical curves for reliability in the wear failure mode (normal curves). Also confidence limits and standard deviation are of interest because they yield a measure of uncertainty due to the number of data points available and their spread about the mean. Figures 7-6 through 7-12 show these parameters. Lack of data is particularly noticeable in the case of interstage seal ring lives.

The component lives are subject to question and ultimate revision due to the application of judgment factors. Certainly it would be more accurate if an absolute clearance increase could have been used rather than a relative one, and this procedure will be followed for the final report. To check the effects of using absolute clearance, a reliability curve for wear rings was generated on the assumption of a 0.020 inch increase in clearance (double the prototype design clearance), and all data were weighted on this basis. It was found that mean life in this case was 22,900 hours, with 95% confidence limits from 20.2 to 25.6 thousand hours and standard deviation of 11,600. The comparison of field mean lives based on absolute wear does not alter either the water quality factors or the general picture of life versus frequency of unit starts, nor does it indicate a re-evaluation of materials is needed.

A summary analysis was made to find out if the 31,500 life predicted for the four-stage prototype on the basis of field experience was more reasonable than the 22,900 wear ring life based on original prototype clearance. An efficiency difference of one percent is estimated to cost about $28.20 per hour for 9.4 MW of power at three mils per kilowatt-hour (Ref. 11). For each of sixteen pumps in the single lift concept, the cost is $1.76 per hour per unit per one percent loss in efficiency. On the basis of leakage equations used in this study, it was determined that wear ring efficiency loss for the four-stage prototype would be about 0.06% initially. The loss would increase by about 0.06% every 22,900 hours when clearance had increased another 0.020 inches. There are four wear rings in the prototype but experience has shown that the suction ring wears at a greater rate so the

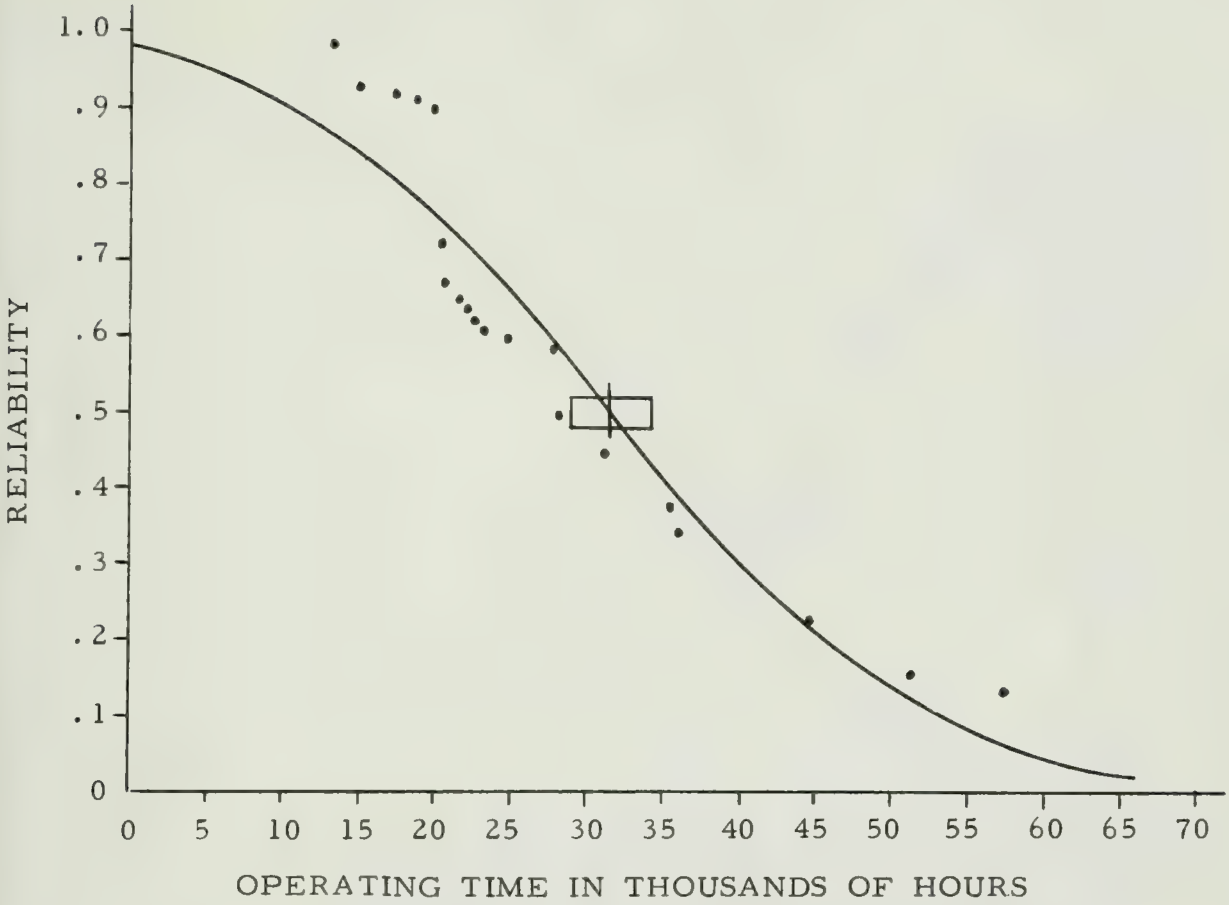


FIGURE 7-6 PREDICTED WEAR RING RELIABILITY

Mean Life = 31.5 k Hours
95% Confidence Limits = ± 3.8 k Hours
Standard Deviation = 16.4 k Hours
Bar Denotes Confidence Interval
Dots Denote Observed Function

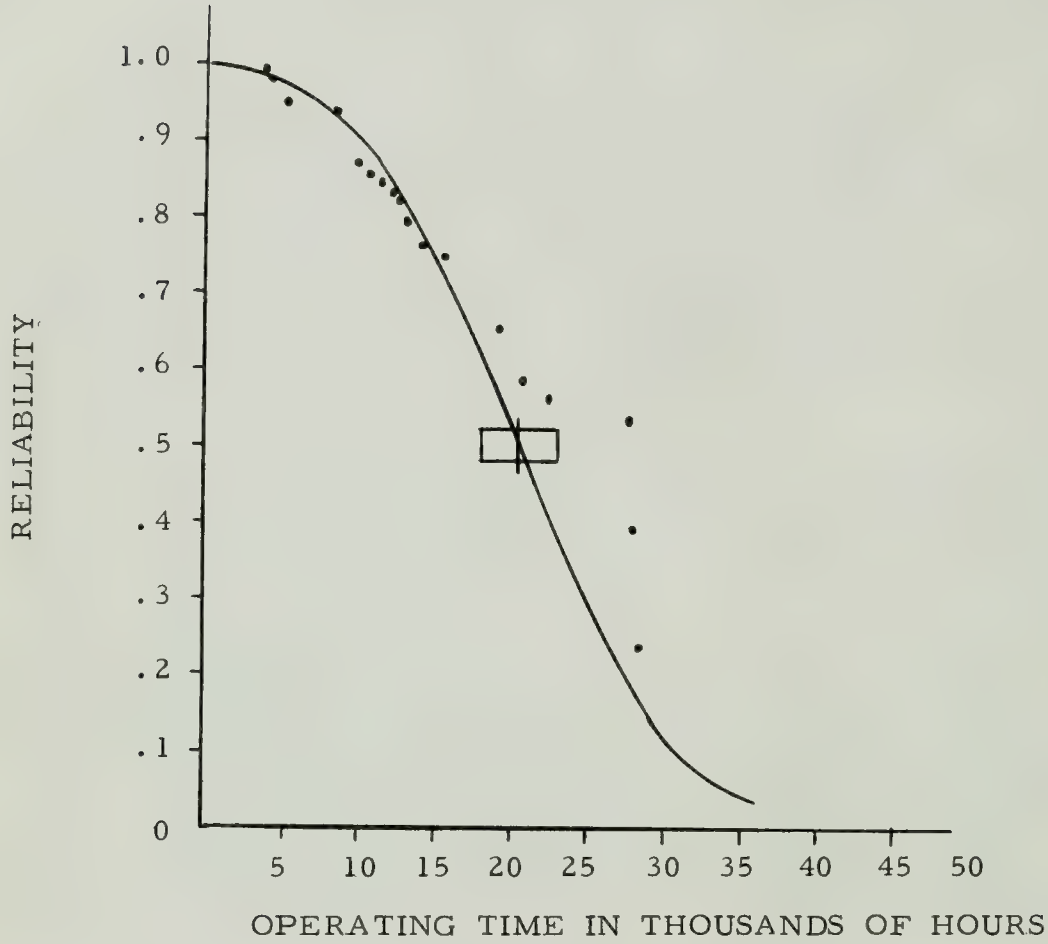


FIGURE 7-7 PREDICTED BALANCE RING RELIABILITY

Mean Life = 20.4 k Hours
95% Confidence Limits = $\pm$ 2.5 k Hours
Standard Deviation = 8.2 $\overline{k}$ Hours
Bar Denotes Confidence Interval
Dots Denote Observed Function

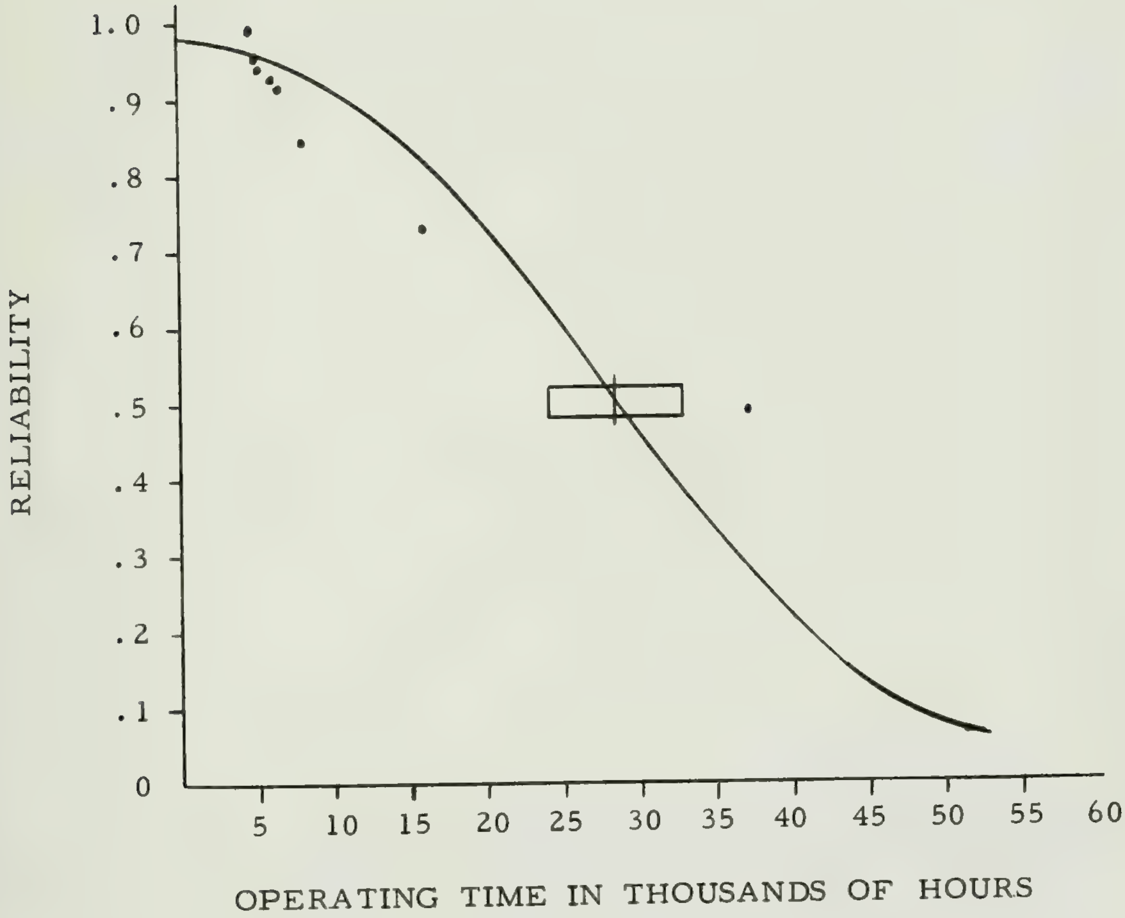


FIGURE 7-8 PREDICTED IMPELLER RELIABILITY

Mean Life = 28.5 k Hours
95% Confidence Limits = $\pm$ 4.5 k Hours
Standard Deviation = 13.8 k Hours
Bar Denotes Confidence Interval
Dots Denote Observed Function

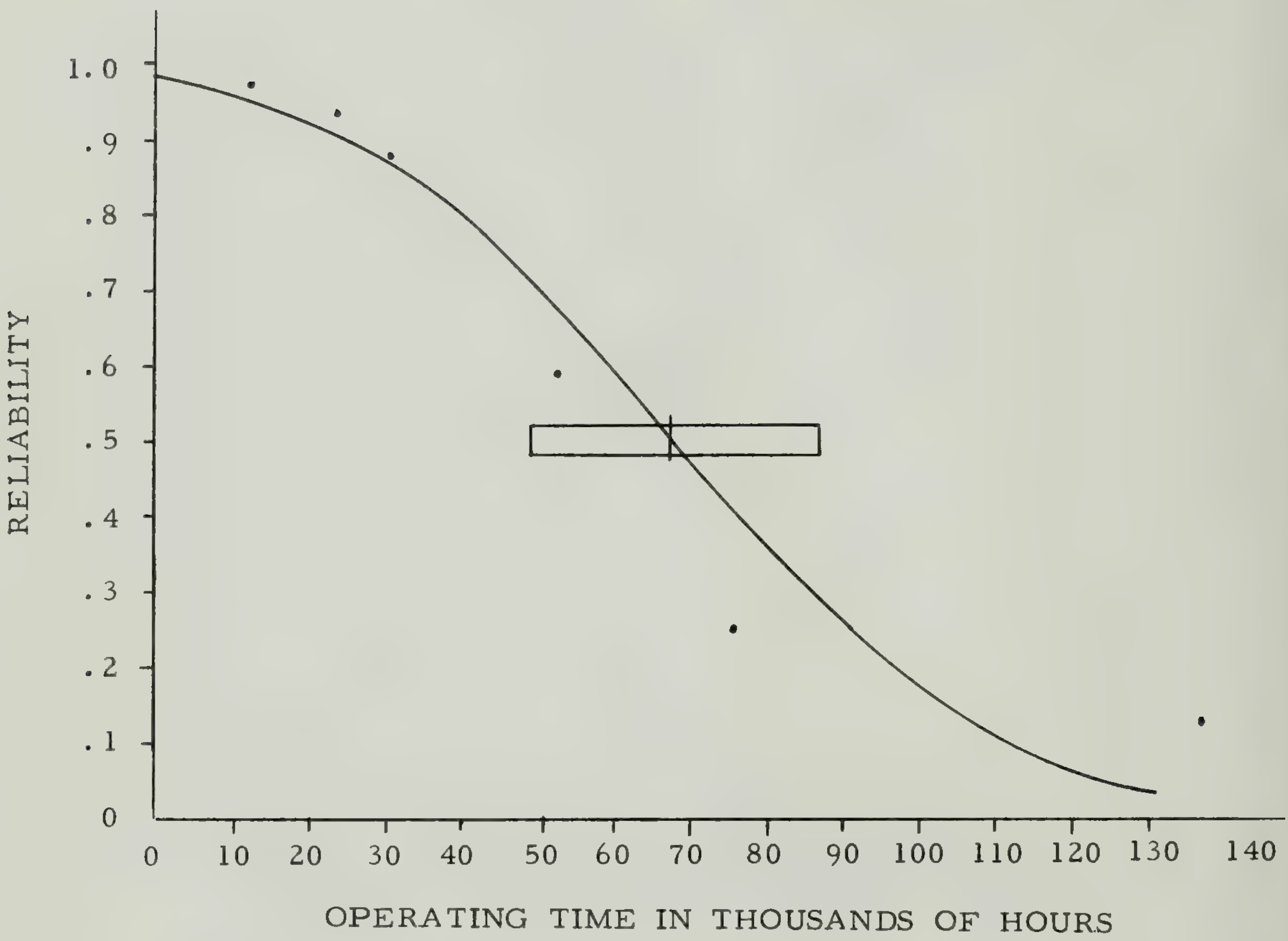


FIGURE 7-9 PREDICTED INTERSTAGE SEAL RELIABILITY

Mean Life = 67.8 k Hours
95% Confidence Limits = ± 18.9 k Hours
Standard Deviation = 33.4 k Hours
Bar Denotes Confidence Interval
Dots Denote Observed Function

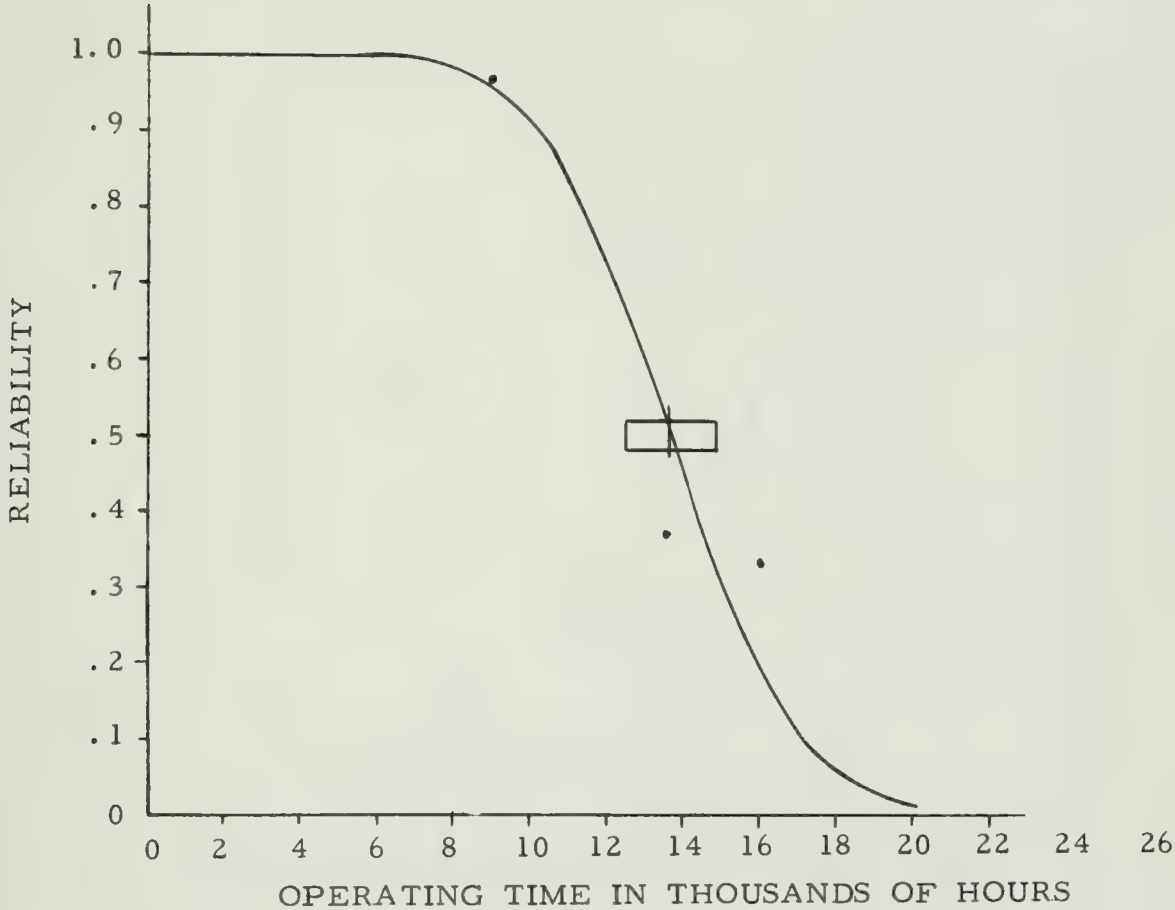


FIGURE 7-10 PREDICTED FIXED BUSHING SHAFT PACKING RELIABILITY

Mean Life = 13.7 k Hours
95% Confidence Limits = $\pm$ 1.2 k Hours
Standard Deviation = 2.6 k Hours
Bar Denotes Confidence Interval
Dots Denote Observed Function

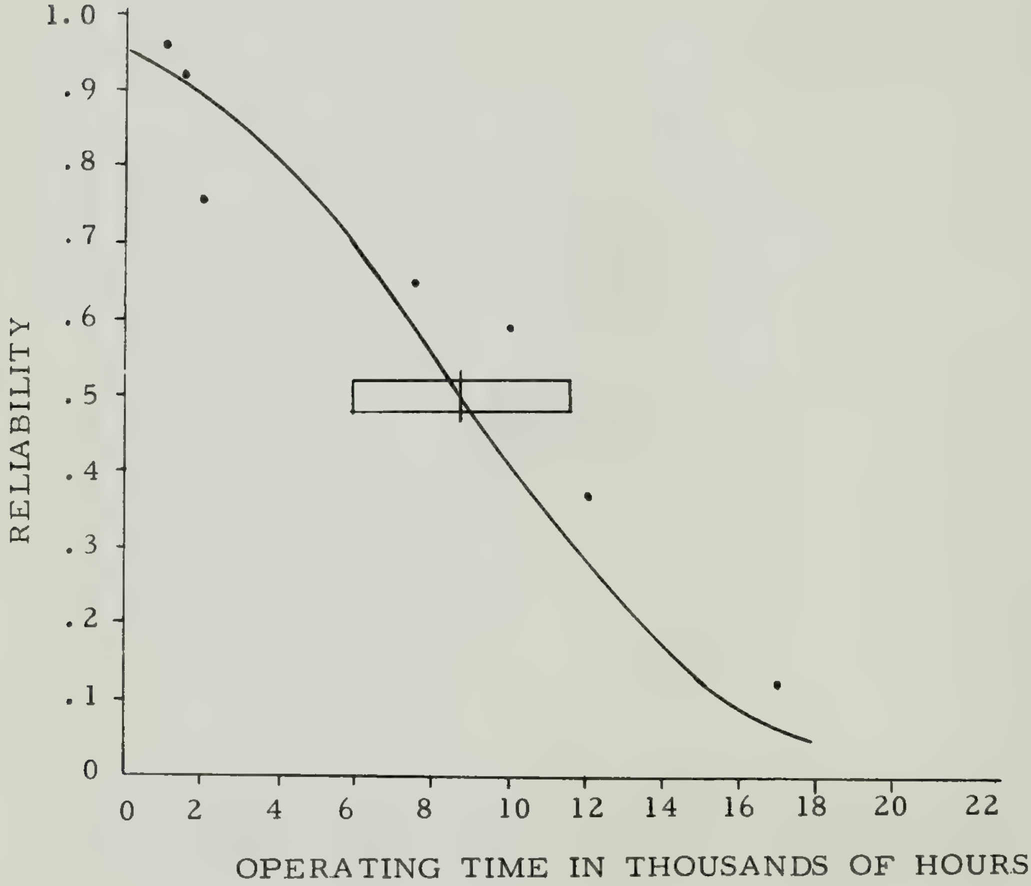


FIGURE 7-11 PREDICTED CARBON RING SHAFT PACK RELIABILITY

Mean Life = 8.7 k Hours
95% Confidence Limits = $\pm$ 2.8 k Hours
Standard Deviation = 5.3 k Hours
Bar Denotes Confidence Interval
Dots Denote Observed Function

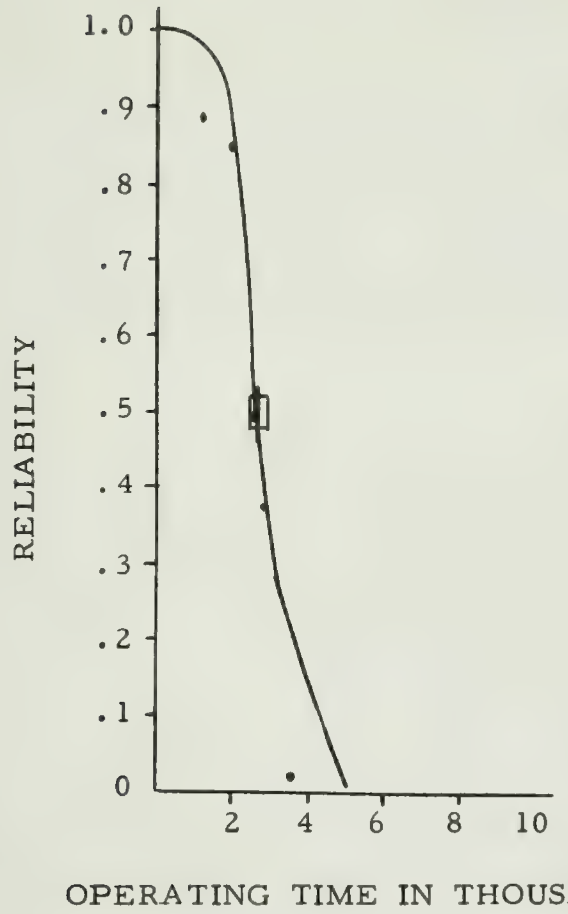


FIGURE 7-12 PREDICTED SOFT PACKING RELIABILITY

Mean Life = 2.7 k Hours
95% Confidence Limits = ± 0.2 k Hours
Standard Deviation = 0.6 k Hours
Bar Denotes Confidence Interval
Dots Denote Observed Function

other three rings might each be assumed to lose 0.75 times the efficiency lost by the suction ring in the same period. Interstage seal ring life was computed on the basis of about the same efficiency loss as the wear ring but it takes twice as long to reach this point according to component lives so the three seal rings will each be considered as losing 0.5 of what the wear rings lose in efficiency at any time. Then the portion of one percent efficiency lost by one suction and three downstream wear rings plus three interstage seal rings is (4.75)(.06) (operating hours)/22,900 hours. Cost of each one percent loss is $1.76 per hour times operating hours and the product of average loss and cost per loss should equal cost of the overhaul if it is done on an economic basis (power cost only).

Cost of an overhaul is based on very fragmentary information. The price of two sets of seven foot diameter, smooth surfaced, stainless steel rings, in one instance, was found to be $8,125. For this analysis it was assumed that wear rings cost $4,000 per set and seal rings which will be made of less expensive materials cost $1,000 per set. Labor costs of $5 per hour for five men over an overhaul period of 250 hours will be assumed in addition to material costs for a total of $25,250.

It is found that cost of lost efficiency will equal cost of an overhaul in 48,000 hours for the single lift prototype pump. Predicted component lives are seemingly shorter than they may be when operations at Tehachapi commence, but this is preferable because of uncertainties involved in estimating water quality, velocity, material resistance and probability of adequate submergence. The important consideration for this report is that comparison of lift concepts is realistic. Similar rough analysis of economic conditions were, therefore, applied to the other lifts to make the comparison.

The two lift prototype has two suction wear rings, two aft wear rings, and two interstage seal rings for a total of 4.5 efficiency losses as compared to 4.75 in the one lift. Because of lower horsepower per unit the cost per one percent loss goes down from $1.76 to $1.56 per hour. It will be assumed that initial leakage in the wear rings for the two concepts is similar which seems reasonable since they both experience equal head per stage. However, the total capacity-head product for the two stage unit is slightly smaller than that for the four-stage, so efficiency loss should be (313)(1951)/(557)(975) or 1.12. It will be assumed that initial ring clearance is 0.020 inches for the two-lift unit and its ring diameter does not differ greatly from that of the single-lift. Then the same 22,900 hour period as used above should result in a loss which is double the initial loss for the wear rings. Cost of overhaul is calculated on the same price scale as above

with the result that it is $24,250 and economic repair time is 48,500 hours for the two-lift.

The three-lift prototype has a cost of only $1.04 per one percent loss per hour because of its low power motor, and its overhaul consists essentially of replacing 1.75 suction rings. However, if the same initial leakage per ring is again assumed, the percentage of efficiency loss per .020 inches increase in clearance is 2.22 times what it is for the single lift prototype because of difference in head per stage and total head and capacity. Overhaul costs are again assumed $4,000 per ring plus $5 per hour for five men over a shorter period of 230 hours for a total cost of $13,750. Economic life for these rings is then 50,900 hours.

Two qualifications must be put on the above relative life comparisons. The first is that overhaul time for the multistage pump might be considerably longer than assumed so that costs of overhaul may be more, and time between overhauls would be extended proportionately. Second, it must be pointed out that assuming the same leakage for the single stage pump rings as for the others is giving it an unrealistic advantage in view of the fact that head is about one third higher. Nevertheless, it is preferable that repair time be dictated on an economic basis rather than a "rule of thumb" and it is apparent that single stage pumps in the United States are allowed to suffer a greater extent of wear between overhauls than their multistage counterparts in Europe. Thus, concept effectiveness is calculated on the basis of the above rough estimates for "case 4" under the tabulations. Here also, balance ring overhaul was calculated on the basis of 0.57% initial efficiency loss as predicted by Allis-Chalmers/Sulzer and twice this loss was assumed to take place at 14,800 hours rather than 22,900 hours for reasons of increased water velocity. The price of the labyrinth was assumed to be five times that of the wear ring or $20,000 because it is built to withstand about five times as much pressure. Overhaul time of 100 hours is multiplied by five men at $5/hr. for an overhaul cost of $22,500. The balance overhaul might profitably take place every 25,700 hours then, but this was cut to 24,000 to make it coincide with general overhaul every second time. Stuffing box lives were adjusted for all lift concepts to make every third replacement coincide with general overhaul, again giving an unmerited advantage to the two and three lift systems in view of contemplated higher submergence pressures involved.

Relative ratings of lift concept effectiveness were not changed materially by awarding the two and three lifts extended life for their rings and shaft packing. This fact, in addition to physical advantages assumed and the lack of confirmed economic data, made it seem that component lives as outlined in the previous selection and backed by common practice and experience are probably the most valid at the present time.

The velocity weight factors will probably be a source of dispute and discussion but the data indicates that they are definitely a factor and that the importance of these weighting factors may have been slightly underestimated rather than overestimated. Witness the following listings of v^2 factors as found for the four stage pump and raw data on the failure times for the plants involved (Table 7-IV).

It may be noted in Table 7-IV that higher velocity factors are quite generally accompanied by shorter times to repair. Water quality, materials, judgment and other factors, of course, enter the picture, but Grand Coulee and Hausern, for example, presumably both have good water. Water velocity certainly appears to contribute to wear if it is not the primary factor involved. Field data corroborates results of laboratory tests. Water velocity effects will be easier to judge when wear tests are completed.

Preliminary results from three initial wear tests support the contention that v^2 is on the conservative side for comparing mean lives. On the basis of tests with 500 and 1000 feet of head and the accompanying flow rates, it appears that wear is proportional to some power of velocity ranging from 2.4 to 10.0 for seven samples with one notable exception. The exception is chrome plate which showed no increase in wear with increase in head on the basis of just two tests.

First wear test results indicate that material weight factors may also be valid. Wear of 13% chrome steel is found to be seven times that of chrome plate, while the Stauffer sand test listed eight relative to a softer 13% chrome sample. The 13% chrome is also found to be 2.5 times more resistant to wear than bronze, which was the factor used from the sand tests.

The material weight factors used are also supported by the data. Certainly, no case is evident in which chrome steel was 170 or 34 times more resistant to wear than cast steel or cast iron, respectively, as suggested by boiler feed pump tests. In fact, when turbine bronze and cast iron balance rings at Motec were switched to turbine bronze and 13% chrome the ring life increased only from 1,400 to 1,800 hours and then dropped to 900 hours in a year because of increased sand content in the water. Cast steel also holds up fairly well in most applications where water is not corrosive as indicated by material weight factors used. These factors will be improved in accuracy by the wear test results.

The water quality factors were the most loosely defined because water quality itself is ill-defined. There is no good wear index which may

PLANT	WEAR RINGS		IMPELLERS	
	v^2 Factor	Unweighted time to repair, hours	v^2 Factor	Unweighted time to repair, hours
Arolla	2.24	1,100	1.23	1,100
Ferpecle	2.06	1,700		
		1,400		
Motec	1.30	5,000		
		2,000		
Grand Coulee	1.27	11,000		
Rio Tuy	1.18	2,000	0.57	6,500
Stafel	1.65	3,300	0.98	2,000
Tracy	0.87	30,000		
Hausern	0.77	60,000	0.40	10,000
Herdecke	0.54	30,000		

TABLE 7-IV VELOCITY WEIGHT FACTOR AND MEAN LIFE COMPARISONS

be applied to water but, as illustrated at Motec, it is a factor which must be considered. It is hoped that wear test results combined with water analyses at Tracy will provide a clearer picture of the dominating wear mechanisms in water. Results of the first 250 hour wear test, if extrapolated and weighted for water velocity and materials, indicates a .020 inch clearance increase in 50, 000 hours for the rings but actual experience at Tracy Pumping Plant belies this preliminary indication.

A study of the times to failure in the raw data shows it is necessarily top heavy with short times due to the fact that the newer plants of greatest interest have not suffered failures as yet unless they have high velocity or poor water or both. This is the best argument in favor of the weighting process as a whole if realistic predictions are to be made. For instance, mean impeller life from raw data alone is about 10, 000 hours but when they are properly weighted for Tehachapi conditions the mean time becomes about 28, 000 hours.

Finally, a defense should be made for the calculation of component life for the one and two stage prototypes on the basis of four stage life weighted by velocity squared factors. It appears, first of all, that this would be the only variable which distinguishes between the operational characteristics of one stage, two stage, and four stage pumps used in the same application. Secondly, an attempt has been made to make every phase of this study reflect common pumping practices for if it did not, field data would not really be applicable in making reliability predictions.

The single stage pump is not generally used in high head situations so that its ring clearances are generally smooth and straight for ease in machining and fitting. Leakage is generally low because of low head but it usually represents a higher percentage of the power lost than in multistage pumps on which designers tighten up on leakage rates to keep efficiency up despite more rings, more disk friction and sometimes balance device losses. In general then, it seems reasonable to assume that water leakage velocity in the single stage prototype will be higher than that in the multistage prototypes because its head per stage is higher, its inherent efficiency advantage will allow greater comparative clearance tolerances and its lack of a history of higher head applications may penalize attempts to improve on straight, smooth bores.

The velocity squared calculated for wear rings of the two stage prototype is only slightly greater than that calculated for the four stage pumps. This reflects in general the close proximity of v^2 factors found for

these types of pumps in field studies. The usual assumption is that double flow pumps will exhibit slightly higher efficiencies than multistage single flow pumps for the same total head and capacity. This indicates that the double flow pump could probably afford a little more ring leakage and generally does.

It is probably quite true that a designer could alter the length, roughness, and configuration of a ring to make it match any other in water velocity despite differences in head and rpm. For this reason, lift concept effectiveness is calculated on the assumption that velocity squared does not vary for the components of the three prototypes and it will be seen that results do not differ greatly from those based on the above mentioned weighting process (case 2 of Table 7-VI).

3. Component Material and Design

The only materials on which there is sufficient information to judge their relative merits are the cast iron-bronze and bronze-bronze ring combinations. The former appears to be 1.5 to 2 times more resistant to wear than the latter. The chrome-bronze ring combination appears to be about equal to but probably better than the cast iron-bronze ring. Most common, by far is the cast iron-bronze which gives quite good wear in most cases.

Isolated cases of ring materials which outlast the cast iron-bronze average indicate that a chrome-cast iron combination holds most promise followed by manganese steel. Stainless steel ring combinations of different hardness are being field tested at several installations but their operating lives are too brief to draw any conclusions. The 1020 and 1040 steel rings assumed in this study are in use at Lewiston and Hiwassee. Hiwassee has run about 6,000 hours without trouble. Lewiston changes its lower rings about every 3,700 hours but the upper rings of the same material show no wear when replacement occurs.

No material recommendations will be made as a result of this study inasmuch as choice of materials should not be limited to those which have been investigated in the field. Pump rings are too long-lived to field test for best materials. In their technical review of 1961 (Ref. 7-14), Sulzer recommends wear rings having a combination of 13% Cr. steel with a Cr - Ni - Mn - Mo - V steel to avoid seizing and cracking which results when high chrome steels are used. The material tested had 0.75 Cr, 3 Ni and 0.6 Mn.

Field interviews indicate that the important point to stress in wear ring design is that clearances be large enough so that maintenance crews may replace them without undue machining to prevent rubbing. Such machining is time consuming, may lead to rings which are out of round and will cause losses in efficiency. It would seem preferable to choose a prototype with a little less efficiency initially than to find design tolerances untenable at overhaul time and be forced to re-design the impeller to make up capacity lost to ring clearance enlargement. Minimum clearance with which many maintenance men find they can work is about 0.020 to 0.025 inches.

Of course, the best ring design as far as wear is concerned is that which allows the lowest relative water velocity in the clearance. Thanks to a moderate choice of pump speeds the multistage prototype ring designs allow water velocities which are about average for the pumps surveyed, but if space and cost allowed it would be beneficial in the long run to have a greater number or increased length of throttling surfaces.

Interstage seal ring design presents no particular problems. Water velocities through these rings are found to be much lower than through the wear rings. This is partly due to lower head across the clearance but also due to the length which is made available between stages for crossover and narrow clearances which are often used with babbitt and steel seals.

Predicted balance ring leakage for the four stage prototype is such that it appears the balance ring will have to be replaced more often than the wear rings if values assumed in the water velocity equations are realistic. Methods should be explored to bring these mean lives closer together by increasing wear tolerance or length of throttling surface, and so forth. A saving in repair time would probably result.

Three types of shaft packing other than the fixed metal bushing were considered in the study because it seemed possible that any of the four might be used. Soft pack may be used for moderate pressures and peripheral speeds without excess heat or wear (Ref. 7-14). Its mean life was found to be about 2,700 hours with about 8 hours repair time. Carbon ring packing life was calculated to be 8,700 hours which is comparable to the 8,000 hour average life found in the carbon rings for boiler feed pumps (Ref. 7-14). These packings may also be repaired in a short time compared to fixed metal bushing types. However, there have been instances of seizing with carbon rings. The mechanical seal employing self adjusting carbon seals has operated successfully at Vianden for a maximum of about 3,600 hours. It is still considered a "delicate organ whose behavior is difficult to predict" (Ref. 7-14)

With reference to impeller life, it should be emphasized that wear in these predictions was considered due to erosion and corrosion in the absence of cavitation. If cavitation is present, wear may proceed at a rate comparable to the 6th or 7th power of the fluid velocity rather than the second power as assumed here (Ref. 7-15).

4. Maintenance Schedule Tabulation

The table (7-V) shows estimated maintenance schedules for each prototype pump in the three lift concepts. It is assumed that repairs are made simultaneously at each plant.

5. Validity of Maintenance Schedules

Unfortunately the repair times in the maintenance schedule are based on less information than was used to compute times between repairs. Because of uncertainty in repair times on the basis of available data, lift concept effectiveness was calculated using repair time and schedule from ref. 7-11. The schedule is probably realistic in that it lists balance ring replacement separately from overhaul time for the four stage pump. Experience has shown this to be quite common.

6. Lift Concept Effectiveness Tabulation

Table 7-VI gives the essential information which went into the calculations of concept effectiveness. The items, by column number, are explained verbally here.

(1) Mean time between failures is average time between unplanned outages according to data from MWD aqueduct plants which was adapted for each plant of the three lift concept and weighted for each plant in the other concepts by pump complexity factors.

(2) This column lists the probability that each plant of the lift concept will operate for 95% of any year (8350 hrs. operation required by 5% overcapacity) without suffering an unplanned outage. The exponential form is used here because it best describes the constant hazard rate associated with unplanned outages and it is verified in plotting data from individual plants.

ONE LIFT CONCEPT					TWO LIFT CONCEPT				THREE LIFT CONCEPT			
Pump hours	Repair hours	Repairs * 1	2	3	Pump hours	Repair hours	Repairs * 1	3	Pump hours	Repair hours	Repairs * 1	3
14,000	50	x			13,900	50	x		12,900	50	x	
21,000	100		x		27,800	50	x		25,800	50	x	
28,000	50	x			31,200	250		x	29,000	230		x
31,500	250			x	41,600	50	x		38,700	50	x	
42,000	100	x	x		55,500	50	x		51,600	50	x	
56,000	50	x			62,400	250		x	58,000	230		x
63,000	250		x	x	69,300	50	x		64,500	50	x	
70,000	50	x			83,200	50	x		77,200	50	x	
84,000	100	x	x		93,600	250		x	87,000	230		x
94,500	250			x	97,000	50	x		90,200	50	x	
98,000	50	x			111,000	50	x		103,000	50	x	
105,000	100		x		124,800	250	x	x	116,000	230	x	x
112,000	50	x										
126,000	250	x	x	x								

* Repairs: 1 = stuffing box; 2 = balance rings; 3 = overhaul

TABLE 7 - V PREDICTED LIFT CONCEPT MAINTENANCE SCHEDULES

LIFT CONCEPT	MEAN-TIME* BETWEEN FAILURES (1)	$e^{\frac{-8350}{(1)}}$ ** (2)	TIME AVAILABLE FOR OUTAGES PER YEAR: Planned* (3)	TIME AVAILABLE FOR OUTAGES PER YEAR: Unplanned* (4)	UNPLANNED* REPAIR TIME PER REPAIR (5)
One Lift	21,100	.672	113	297	105
Two Lift	22,300	.687	94	158	100
Three Lift	31,000	.766	95	105	102

LIFT CONCEPT	REDUNDANCY FACTOR $\sum_{r=0}^{r=\frac{(4)}{(5)}} \frac{\left[\frac{(8350)}{(1)}\right]^r}{r!}$ (6)	OPERATIONAL RELIABILITY P_r $\left[(2) \times (6)\right]^{\text{lift }\#}$ (7)	MEAN* UNPLANNED OUTAGE TIME PER YEAR (8)	OPERATIONAL READINESS P_{or} (Availability) (9)	CONCEPT EFFECTIVENESS P_{ce} (7) x (9) (10)
One Lift	1.48	0.997	41	.982	.979
Two Lift	1.41	0.970	75	.980	.950
Three Lift	1.27	0.979	82	.979	.902

* All times listed in hours

** System Assumed 100% Reliable for Planned Outages

TABLE 7-VI LIFT CONCEPT EFFECTIVENESS TABULATION

(3) The mean annual planned outage time per concept is given as derived from Table 7-V predicted schedules. It is the needed annual operating time divided by total accumulated operating hours per concept as listed in the Table and multiplied by total repair time in the total accumulated hours for a cycle of repairs.

(4) Time available to repair unplanned outages per plant is time left in the year after mean annual planned repair and operating time are subtracted from total hours in the year. This time is then divided by two for the two lift concept and three for the three lift concept to make repair time for each plant independent of the others in the concept because time taken in one is taken in all.

(5) Unplanned repair time per repair is derived for the three lift concept from mean time between failures (col. (1)) and 0.31% mean annual unplanned outage time per plant as found in the DWR unplanned outage analysis. This time is extrapolated to the other lift concepts by considerations of time taken for each additional repair indicated by pump complexity.

(6) The summation which results in a redundancy factor represents a count of the ways in which each plant of the concept might succeed as limited by the number of repairs allowed by dividing column (4) by column (5). Each added term represents the product of probabilities that "r" failures take place and "r" repaired units succeed after repair. The product is used to express the fact that failure must occur before repair and renewed successful operation is assumed.

(7) Operational reliability is the total sum of all ways a plant may succeed taken to a power equivalent to the number of plants in the concept because the plants must succeed in a series for unplanned outages.

(8) Mean unplanned outage time per year is operating time per year divided by mean time between failures (1) as multiplied by unplanned repair time per repair (5) and number of plants per lift concept.

(9) Operational readiness is annual hours minus the sum of columns (3) and (8) divided by annual hours.

(10) Concept effectiveness is the product of operational reliability (7) and operational readiness (9).

Table 7-VII shows the effects of increasing overcapacity on concept effectiveness. It may be noted that two lift overcapacity must be increased to 10% and three lift overcapacity to about 15% in order that they may both have the same concept effectiveness as the single lift with 5% overcapacity. Equal operational reliability may be attained with 10% overcapacity on the three lift and 10% on the two lift.

7. Validity of Lift Concept Effectiveness

The validity of lift concept effectiveness as a comparative index does not appear to be subject to dispute at this time. Neither the reliability nor the availability of individual pumps designed for about the same head per stage, capacity and speed varies enough to be a factor whether the pumps be one, two, or four stage. Where the one lift concept is inherently more reliable than the other two is in the fact that it requires only one system to operate successfully while the others demand success of two out of two or three out of three. If the chances of one system succeeding are x, the chances of three systems succeeding are x^3, because the chances of having a failure are greater.

In the knowledge that some error might be involved in utilizing the assumptions, judgments and techniques which formed the basis for this report, a series of validity tests were conducted to determine whether any logical variation in this basis could lead to changes in the order of concept effectiveness figures or even significant changes in comparative values. All test conclusions confirmed the validity of lift concept effectiveness as listed in this report.

The tests involved detailed calculations of concept effectiveness and its component parts incorporating the following variations:

Test 1: Scheduled outage time was assumed to be 2.7% as presented in ref. 7-11. This represents the portion of time the entire aqueduct is out of operation and available for repairs, inspection, alterations and so forth. For Test 1, it was assumed that 2.3% of the year (5% overcapacity minus scheduled outage time) was available for unscheduled outages. It was assumed that equal times were available for repair for all concepts.

OVERCAPACITY ALLOWANCE	ONE LIFT			TWO LIFT			THREE LIFT		
	P_r	P_{or}	P_{ce}	P_r	P_{or}	P_{ce}	P_r	P_{or}	P_{ce}
2.5%	.9295	.9820	.9128	.7467	.9804	.7321	.6914	.9795	.6772
5.0%	.9974	.9823	.9794	.9704	.9803	.9504	.9214	.9793	.9022
7.5%	.9995	.9826	.9821	.9968	.9809	.9778	.9819	.9800	.9623
10.0%	1.0000	.9829	.9829	.9997	.9812	.9809	.9981	.9802	.9783
15.0%	1.0000	.9835	.9835	1.0000	.9816	.9816	1.0000	.9807	.9807
20.0%	1.0000	.9839	.9839	1.0000	.9820	.9820	1.0000	.9811	.9811
25.0%	1.0000	.9844	.9844	1.0000	.9824	.9824	1.0000	.9815	.9815
30.0%	1.0000	.9848	.9848	1.0000	.9828	.9828	1.0000	.9819	.9819

LEGEND: P_r = Reliability
P_{or} = Availability
P_{ce} = Concept Effectiveness

TABLE 7-VII OVERCAPACITY EFFECTS ON LIFT CONCEPT EFFECTIVENESS

Test 2: Reliability here dictates that rings and stuffing boxes in the one and two stage pumps be designed to allow no greater water velocity factors than found in the four stage pump. This puts all lift concepts on an equal footing as far as component life is concerned, but penalizes the one lift concept slightly by assuming separate repair of balance labyrinths.

Test 3: Test three assumes that repair schedules as drawn up in ref. 7-11 are applicable. Component life is improved possibly through use of new materials so that overhauls are performed every nine years. Overhaul time is 480 hours for the one and two lift concepts and 240 hours for the three lift. Sixteen hours are added for inspections and lesser maintenance.

Test 4: Since the two and three lift prototype has fewer parts to replace at overhaul and since equal losses in efficiency cost less in power, it is postulated that time between overhauls based on economics for these lift concepts may exceed that for the single lift. Estimates of repair times are derived in the section on validity of component life predictions. Wear rates are assumed equal in the prototypes as in Test 2, providing an unmerited advantage to the three lift concept in view of its head per stage.

Test 5: This test compares the lift concepts with regard to a combined distribution of planned and unplanned outages rather than assuming 100% reliability for planned outages. It utilizes some of the techniques recommended below for incorporation in the final report on absolute reliability.

It might be well to recall here the character of the study basis for the results given in this report. Generally patterned after common practice in design and field maintenance, this report assumes the rings and stuffing boxes in the one and two stage pumps will wear at slightly greater rates because of increased pressures across the clearances and decreased efficiency demands as discussed previously. It also assumes separate repair of the balance ring for the single lift concept. Tehachapi is considered independent from other aqueduct operations.

For the final report, it is proposed that all times between maintenance action be included in determining operational reliability whether they be scheduled or unscheduled. Of course, the times between pump repair outages will be those from the data as weighted for Tehachapi conditions including economic clearance tolerances. The times between outages for motor and valve maintenance may also be weighted for Tehachapi parameters if correlations can be found.

Times for maintenance actions will be plotted on one curve whether planned or not. Again the best fit distribution will be used to get mean repair time and probability that a repair is completed in any time interval of interest. This probability may be treated as a switching reliability in a standby redundancy situation as described by:

$$r(t) = \exp(-t/m)(1 + p_1 t/m + p_2 t^2/2m^2 + p_3 t^3/6m^3 + p_4 t^4/24m^4 + \dots) \qquad (7\text{-}25)$$

where $r(t)$ = reliability of a unit at time "t"

m = mean time between outages

p_1 = probability of completing one repair in time allotted

p_2 = probability of completing two repairs in time allotted

p_3 = probability of completing three repairs in time allotted (and so forth)

Time allotted for each repair is time not needed because of over-capacity divided by number of repairs. The above summation would be carried on only as long as necessary to determine the limiting value. Equation (7-25) really applies only to the exponential case and must be modified if other distributions are more fitting.

Probability that the concept is ready to operate at any time of year it is needed should be calculated from

$$P_{or} = P_a - P_a P_s + P_s \qquad (7\text{-}26)$$

where P_{or} = operational readiness

P_a = operational availability

P_s = standby readiness

The form of equation used here is the result of an "or" situation. That is, the system can operate because it is operationally available <u>or</u> it is in a state of standby readiness. Operational availability is percent of

calendar time when the system is operating or ready to operate if demanded because it does not need repair. It is operating hours divided by operating hours plus some factor times outage time. The factor is used to reflect the fact that availability when demanded may be increased by repairing the system when it is not in demand. The standby readiness is percent of calendar time when the system is ready to operate if demanded because operation has not been needed. A factor may also be used here which reflects the chances that a unit or a whole plant may be forced to submit to maintenance action despite its standby mode. As long as repair time is not included in the standby, the operational availability and standby readiness should be mutually exclusive and equation (7-26) should be applicable.

Other areas in which validity of the lift concept effectiveness will be improved for the final report include data gathering and computation. Emphasis will be more on absolute accuracy of predictions than comparative.

Data gathered will be improved in accuracy and detail. Operating hours to repair and component condition at repair will be clarified. Much more data on costs of repairs and times of repairs will be gathered so that repair time distributions may be formulated and repair cost-effectiveness analyzed. More information on maintenance of motors, valves, bearings, and switchgear will be procured as well as detailed accounts of failure modes involved. Designs, clearances and water qualities will be more accurately determined.

Judgment weighting factors will be placed on absolute bases for the various lift concepts. If possible, a preventive maintenance program for valves, switchgear and motors will be developed with respect to water hammer, voltage, insulation heating and number of starts. Economic and engineering analyses will determine best tolerance specifications based on final prototype designs and efficiency data.

Applicable water velocity factors will be refined by information from the rotary and stationary wear tests. Factors adopted will be verified by closer scrutiny of field experience.

Materials suitable to design, maintainability and budgetary considerations will be chosen for prototype components. The wear test will help to determine comparative operating lives.

Water quality weighting will be placed on a more valid basis by wear test results and reexaminations of field experience in the light of more

accurate techniques and data. Evaluation of unit starts and settling of solids will also be put on firmer ground.

Successful implementation of the above propositions will make concept effectiveness reflect the real picture of Tehachapi's strengths and inadequacies in so far as is possible without detailed studies of upstream operations. It may point out needs for spare pumps or increased overcapacity to be sure of reliable water delivery and it will present a clearer overall picture of comparative lift concept advantages.

G. APPENDIX I---BIBLIOGRAPHY

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CHAPTER 8

METHODS OF ESTIMATING THE EFFICIENCY OF HYDRAULIC MACHINES FROM MODEL TESTS

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NOMENCLATURE

a = dimensionless measure of kinetic fraction of total head loss

c = dimensionless coefficient in McDonald formula

C_i = typical kinetic (friction) loss coefficient

d = diameter of pipe

D = diameter of pump impeller or turbine runner

D_o = eye diameter of pump impeller

f = friction factor; dimensionless coefficient in McDonald formula; function of hydraulic efficiency in Rütschi formulae

F = friction coefficient in Biel formula

g = acceleration of gravity

h = total head loss in pump or turbine

h_k = kinetic head loss in pump or turbine, i.e. loss proportional to second power of velocity

h_f = head loss due to surface friction in pump or turbine

H = net head for pump or turbine

i = subscript to represent a typical value of several like quantities

k_F = measure of surface roughness introduced by Fromm and used in Pantell formula

ℓ = length of flow passage, flat plate, etc.

m = exponent; hydraulic radius

n = exponent

n_s = specific speed of pump or turbine

N = shaft speed of pump or turbine, revolutions per minute

o = Subscript to designate the point of maximum efficiency unless otherwise specified

Q = capacity of pump or discharge of turbine

Re = Reynolds number

u = $\pi DN/60$ = peripheral speed of pump impeller or turbine runner

v = average fluid velocity

V = average fluid velocity

x = variable; measure of length parallel to direction of flow

y = variable

δ = dimensionless ratio of total head loss to net head

δ_f = dimensionless surface friction loss parameter

δ_k = dimensionless kinetic loss parameter

Δ_1 = error defined by Eq. (28)

Δ_2 = error due to assuming $\eta_h'/\eta_h = 1$ in formulae for predicting pump efficiency

ϵ = linear measure of absolute surface roughness

η = overall efficiency

η_h = hydraulic efficiency

η_m = mechanical efficiency

η_v = volumetric efficiency

ν = kinematic viscosity of fluid

ρ = mass density of fluid

$\sum_i$ = summation of i terms of like character

ϕ = ratio of hydraulic efficiencies in Eq. 3b; function

Primed (') quantities refer to a model pump or turbine

Unprimed quantities refer to a prototype pump or turbine.

METHODS OF ESTIMATING THE EFFICIENCY OF HYDRAULIC MACHINES FROM MODEL TESTS

A. REPORT IN BRIEF

1. Purpose

The purpose of this report was to recommend suitable methods for estimating the probable efficiencies of the Tehachapi pumps from tests of scale models.

2. Literature Search

The American, British, German, Swiss, and Japanese literature was surveyed and more than fifty references obtained. These were studied with considerable care including checking of numerical work wherever possible. The validity of approximations used in several formulae were tested as shown in Table 1 and Figs. 1 and 2. Table 2 lists more than forty formulae for estimating the efficiency of pumps and turbines from the results of tests of scale models. Fig. 3 and Table 3 show a comparison of several formulae some of which have been used for both pumps and turbines.

3. Discussion and Conclusions

From a broad viewpoint, all turbines or pumps should fit some comprehensive pattern so that information on one machine could be used to predict the performance of any other. Until such time as this can be done, it has been customary to define a model as a comparatively small machine which is, for the most part, geometrically similar to a much larger machine called the prototype.

Most of the models have been built and tested by the firms that have designed and built the prototype machines and the details of the model tests have remained their property. Individual manufacturers have been understandably reticent to release the volume of numerical data necessary either to test formulae already proposed or to develop better ones. Several formulae, such as those due to Moody and to Ackeret, were developed by individuals having access to the records of a particular manufacturer. Thus each such

formula was designed to fit the data on which it was based and appears to have done so very well. It is most likely that individual practices in the construction and testing of models has differed so much that no single formula can be expected to have universal applicability. A number of formulae have been proposed on a basis of limited theoretical considerations with few if any numerical data to support them. Moody in particular [22] [1] has warned against placing too much reliance on any formula which has not been substantiated by a large number of tests. [2] It may be concluded that any formula or method for converting the efficiency of a model to that of a prototype machine should have received such substantiation before being applied to an installation as important as the Tehachapi pumps.

Of the many formulae which have been proposed, probably those due to Moody and to Ackeret have had the widest acceptance and are based on the largest amount of data from both model and field tests. The Ackeret formula [3] [35] used the Reynolds number to estimate the surface friction losses in both model and prototype. It is doubtful if any appreciable effect of viscosity can be detected where the fluid is cold water [4] in both model and prototype so that the conversion is based on the geometrical scale ratio and the ratio of model to prototype head. Mühlemann [35] has shown that the Ackeret formula agreed with the results of tests of some 30 to 40 turbines [5] to within about ±2%. These data were all supplied by the Ateliers des Charmilles S.A., Geneva, Switzerland, and may be expected to reflect the individual practices of that company.

The several Moody formulae [9, 17, 25, 29] were changed as time passed. The first formula [9] was similar to the Ackeret but, as experience accumulated, Moody found that the influence of any difference in head between the model and prototype had become negligible for the data available to him [25, 33] and that the exponent of the geometrical scale ratio tended to decrease as the art of constructing and testing models improved. He also pointed out [6] that wide variation might be expected in the results of model tests due to

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[1] Numbers in brackets refer to the bibliography, Appendix I, Section G.

[2] Page 376 of reference [25].

[3] Pfleiderer's 1947 formula [34] was a close approximation to the Ackeret formula. Moody [9] used the same type of approximation for most of his published formulae.

[4] Less than 85°F.

[5] See Fig. 9.

[6] Page 91 of reference [33].

individual practices in construction and testing. Kerr[1] has stated that different laboratories reported efficiencies differing by as much as two percent for the same model with the I. P. Morris laboratory usually reporting the low values. This probably accounted for a large part of the difference in efficiency as computed by the Ackeret and Moody formulae because the Moody formulae were based mostly on the I. P. Morris tests.

Table 3 shows prototype turbine and pump efficiencies as predicted by several formulae. The differences, which in some cases amount to several percent, must in large measure be due to the individual practices in fabricating and testing models because each formula may be assumed to have represented accurately the data on which it was based.

Unfortunately there is no single arbiter of efficiency conversion. The American Society of Mechanical Engineers Power Test Code, PTC 18 - 1949, Hydraulic Prime Movers, makes no mention of efficiency conversion. The Standards of the Hydraulic Institute, Tenth Edition, Second Revision, Page B (VIII) -21, recommends the turbine formula.

$$(1-\eta)/(1-\eta') = (D'/D)^n \tag{1}$$

for use with geometrically similar pumps with $0.26 \leqq n \leqq 0$ depending mostly on the differences in surface roughness and clearances of the model and prototype pumps. In Eq. (1), η is the efficiency[2] and D is the impeller diameter of the prototype. The primes signify corresponding quantities for the model. The International Test Code for Hydraulic Turbines Using Laboratory Models for Acceptance Tests, 1964, recommends that, in the absence of any mutual agreement to the contrary, the Moody formula

$$(1-\eta)/(1-\eta') = (D'/D)^{0.20} \tag{2}$$

be used for geometrically similar Francis turbines. For convenience, η and η' are assumed to cover the hydraulic efficiencies.

The Society of German Engineers (VDI), Centrifugal Pump Code, Acceptance Tests on Centrifugal Pumps, April 1952, [47] recommends

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[1]Page 730 of reference [17], discussion by Kerr.
[2]All efficiencies are decimal values in this report.

$$\eta_h = 1 - (1 - \phi\eta'_h) \left[(D'N')/(DN)\right]^{0.1} \tag{3a}$$

wherein, D' and D are any corresponding impeller diameters and N' and N are the speeds respectively of the model and prototype pumps. The function, ϕ, is given by

$$\phi = \left[1 - (70/D_o^{1.5})\right] / \left[1 - (70/D_o'^{1.5})\right] \tag{3b}$$

wherein, D'_o and D_o are the eye diameters of the model and prototype impellers respectively in millimeters. This recommendation is accompanied by a strong admonition that considerable uncertainty is to be expected in the conversion of model efficiency to prototype efficiency due to the inevitable lack of similarity in surface roughness and clearances. This is a slight variation of a method due to Rütschi [18, 41, 45, 53, 56, 59, 63] who reported that basing values of ϕ on the eye-diameter of the impeller made possible some relaxation in the requirement that the model and prototype have strict geometrical similarity.

No single formula or method for efficiency conversion has received universal acceptance. Eq. (1) which was first recommended by Moody, has been used extensively with turbines and in at least two cases for pumps. Jaski and Weltmer [60] reported that Eq. (1) with $n = 0.25$ was used to predict both the pump and turbine efficiency of the Tuscarora reversible units at Niagara Falls. Johnson and Wachter [65] reported that Eq. (1) with $n = 0.20$ was used to predict both the pump and turbine efficiency of the Smith Mountain reversible units. Byron Jackson Pumps, Inc., in a communication dated August 12, 1964 stated that Eq. (1) with $n = 0.165$ had been found applicable to their single-stage overhung-impeller design. Sulzer Brothers, in a communication dated August 21, 1964, proposed the formula listed under their name in Table 2, but, in a report[1] dated February 22, 1962, it was stated that "-------a rather good correction could also be obtained by using---------" Eq. (1) with $n = 0.20$. J. M. Voith GMBH, in a communication dated August 21, 1964, recommended either the Ackeret or Pfleiderer (1947) formulae (Table 2), for predicting hydraulic efficiencies and noted that the latter predicted slightly higher efficiencies than the former. The foregoing are typical of the divergence of opinion to be found in the literature on the choice of a conversion formula to fit particular circumstances.

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[1] "Determination of Pump Efficiency", by R. A. Strub and H. Canonica.

4. Recommendations

a. It is recommended that the model or models used to represent the final design of the Tehachapi Pumps be made as accurately to scale as possible. This should extend even to the wearing ring clearances although it is recognized that exact modeling of these may not be practicable. The model pump should be tested both with and without a complete model of the inlet passage to ascertain what effect, if any, this passage may have on the pump efficiency.

b. The surface finish of the model impeller and any guide vanes should be at least 63 microinches for all surfaces in contact with the fluid. The surface finish of all other surfaces in contact with the fluid should be at least 125 microinches.

c. The mechanical losses in the model should be determined as accurately as possible and subtracted from the measured input power to aid in estimating the hydraulic efficiency. Both the overall and the hydraulic efficiency should be reported for the model tests.

d. The hydraulic efficiency, η_h, of the prototype pump should be estimated from that of the model by

$$\eta_h = \frac{1}{1 + \left[(1/\eta'_h) - 1\right] (D'/D)^{0.20}} \tag{4}$$

which is the Moody (1942) turbine formula modified to apply to pumps [31, 50]. The modification was brought about by the manner in which pump head and turbine head have been defined. This will be discussed in Parts D. and E.

e. The meager information on mechanical and volumetric efficiencies indicated that an allowance of one-half of one percent for the product of both the mechanical and volumetric efficiencies for the prototype machine is as good a value as can be recommended. (See Part E. 2. and references [8, 16, 35, 47]). It is suggested that the overall efficiency, η, of the prototype pump be estimated from

$$\eta = \frac{0.995}{1 + \left[(1/\eta'_h) - 1\right] (D'/D)^{0.20}} \tag{5}$$

The use of $\eta_m \eta_v = 0.995$ has very little effect on comparisons among several alternate pumps, all of which are large and highly efficient machines.

B. INTRODUCTION

Interest in the scale effect between model and prototype machines appears to date from the early 1880's. The first conversion formula for turbines was published in 1909 by Professor R. Camerer [2, 3,] of the Technical University of Munich, Germany. It included allowances for differences in size and surface roughness of the machines, and in mean velocity and kinematic viscosity of the fluid. Surface friction was evaluated by a formula for pipe friction proposed by Biel [1]. Biel's formula[1] involved the sum of three terms so that the original Camerer formula was not only unwieldy but required evaluation of a hydraulic radius representative of the flow passages in the machine. This latter step required more detailed knowledge of the machines than often was available. Moody [9a, 9b] wrote that the formula was "in fairly good agreement with actual results of tests." During the next fifteen years, Camerer published several simplified versions of the original formula but it is of interest today mainly in that it laid the ground work on which most of the subsequent effort has been based. No attempt to review all of the literature on this subject will be made here. Rather, a discussion of the reasoning which has led to most of the formulae will be given and followed by a selection of typical formulae for both turbines and pumps.

Most of the published formulae have been developed for use with hydraulic turbines and depend, for the necessary numerical values, on the results of turbine tests. Pumps differ sufficiently from turbines to warrant separate consideration but so few numerical data are available from tests of large pumps that turbine test data have been used at least as a guide to the proper numerical values.

C. GENERAL FORMULAE FOR TURBINES

1. Component Losses and Efficiencies

Each unit mass of fluid which flows through a turbine runner gives up ρgH units of energy, wherein ρ is the mass density of the fluid, g is the acceleration of gravity, and H is the turbine net head. Further particulars about H will be given as part of the discussion of one of the Moody formulae. The runner transmits to the shaft most of the mechanical energy in the fluid.

[1] See page 97 of reference [50].

Friction dissipates the remaining energy as heat except for a small portion that appears as kinetic energy in the tail race and which usually is excluded from the turbine head as measured in the field. A small quantity of the fluid that flows through the turbine bypasses the runner by flowing through the clearance spaces and hence does no useful work. The volumetric efficiency, η_v, is introduced to account for it. A part of the power input to the runner is expended in friction in the bearings and seals and does not appear as useful work at the turbine shaft. It is accounted for by the mechanical efficiency, η_m.

The overall efficiency, η, is given by $\eta = \eta_m \, \eta_v \, \eta_h$ (6)

wherein the hydraulic efficiency, η_h, accounts for all the losses between the inlet to the scroll case and outlet from the draft tube which are not covered by η_m and η_v.

2. Separation of Hydraulic Losses

The hydraulic losses[1] can be separated into two groups; those proportional to the squares of velocities and those proportional to powers of velocities other than the squares. Losses in the first category, sometimes called kinetic losses, are typified by the loss at any sudden change in cross-sectional area such as at the exit from a cascade of vanes having thick trailing edges. Those in the second category, usually called frictional losses, are caused by boundary layer phenomena on all of the wetted surfaces.

a. Kinetic Losses

Define a dimensionless kinetic loss parameter for the model by

$$\delta_k' = h_k'/H' = \sum_i C_i' \, V_i'^2 /2g\, H' \tag{7}$$

and for the prototype

$$\delta_k = h_k/H = \sum_i C_i^2 \, V_i^2/2g\, H \tag{8}$$

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[1] Only the case of turbulent flow is considered here, however, this does not exclude the possibility of laminar boundary layers on vanes or other surfaces.

wherein,

h'_k = kinetic head loss in model,

H' = net head for model

$\sum_i$ = Summation of all i terms of like character,

C'_i = Typical kinetic coefficient for model,

V'_i = Any typical velocity for model, and

g = acceleration of gravity for both model and prototype tests.

All unprimed quantities have the corresponding meaning for the prototype turbine. It is customary to assume that the loss coefficients C'_i and C_i are functions only of the geometry of the flow passages. Thus, for geometrically similar machines, each C'_i of the model is identical with the corresponding C_i of the prototype.

A model and prototype are said to operate under dynamically similar conditions when

$$D'N'/\sqrt{H'} = DN/\sqrt{H} \tag{9}$$

wherein,

D = inlet diameter of runner and

N = shaft speed.

For dynamically similar operating conditions it can be shown that $V_i'^2/2g\ H' = V_i^2/2g\,H$. Thus, it follows that $\delta'_k = \delta_k$ for geometrically similar machines operated under dynamically similar conditions.

b. Friction Losses

A dimensionless friction loss parameter for the model is defined by

$$\delta_f' = h_f'/H' = \sum_i f_i' \, (\ell_i'/m_i') \, (V_i'^2/2g\,H') \qquad (10)$$

and for the prototype by

$$\delta_f = h_f/H = \sum_i f_i \, (\ell_i/m_i) \, (V_i^2/2g\,H) \qquad (11)$$

wherein,

f_i = typical friction loss coefficient,

ℓ_i = typical length of flow passage, and

m_i = corresponding hydraulic radius of flow passage

Geometrically similar machines have $\ell_i'/m_i' = \ell_i/m_i$ and, when operated under dynamically similar conditions, $V_i'^2/2g\,H' = V_i^2/2gH$ as before. The friction loss coefficients are, in general, functions of the local Reynolds numbers.

3. The Reynolds Number

The Reynolds number is a dimensionless parameter which, if correctly evaluated, should be a measure of the ratio of the inertia (disturbing) forces per unit volume of fluid to the viscous (stabilizing) forces per unit volume of fluid. It is the product of a linear dimension of the system, a characteristic velocity of the fluid, and the reciprocal of the kinematic viscosity of the fluid. The linear dimension should represent the geometry of the flow passage while the velocity should represent the dynamics of the fluid motion and it is not always easy to select the single quantities which best do this [66, 67]. The Reynolds number usually used to describe flow in a circular pipe is based on the diameter of the pipe and the average speed of the fluid. Reynolds numbers for hydraulic machines have been defined in many ways. Probably the most commonly used linear dimension has been the outer diameter, D, of the rotor for either a turbine or pump. Instead of selecting a particular velocity, it has been customary to use $\sqrt{2gH}$ with $\sqrt{2g}$ treated as a constant for both model and prototype so that it could be eliminated from the computation.

Thus, an assumed typical Reynolds number, could be written as

$$Re = D\sqrt{H}/\nu = ND^2/\nu \tag{12}$$

wherein, ν = kinematic viscosity of the fluid, and Eq. (9) has been used to replace $\sqrt{H}$ by ND.

4. <u>The Dimensionless Loss Parameter Ratio</u>

Define a series of functions of Reynolds number, $\phi_i(Re_i)$ such that

$$f_i = \phi_i(Re_i) \tag{13}$$

Then a ratio of Eqs (10) and (11) for geometrically similar machines operated under dynamically similar conditions yields

$$\delta_f/\delta_f' = \sum f/\sum f' = \sum_i \phi_i(Re_i)/\sum_i \phi_i'(Re_i') \tag{14}$$

The dimensionless loss parameter, δ, for the prototype machine may be evaluated as follows:

Let the kinetic loss parameter, δ_k, be given by,

$$\delta_k = \delta_k' = a\,\delta' \tag{15}$$

wherein, $a = \delta_k'/\delta'$, represents the kinetic part of the total loss for the model. The friction loss parameter, δ_f, is given by

$$\delta_f = \delta_f' \sum_i \phi_i(Re_i)/\sum_i \phi_i(Re_i) = (1-a)\,\delta' \sum_i \phi_i(Re_i)/\sum_i \phi_i'(Re_i') \tag{16}$$

by virtue of the fact that

$$\delta_f' = \delta' - \delta_k' = \delta' - a\delta' = (1-a)\,\delta' \tag{17}$$

The hydraulic losses in a turbine are given by

$$h = h_k + h_f \tag{18}$$

and the hydraulic efficiency by

$$\eta_h = (H+h)/H = 1 + h/H = 1 + \delta \tag{19}$$

It follows from Eqs. (14) - (19) inclusive that

$$\delta/\delta' = (1-\eta_h)/(1-\eta_h') = a + (1-a) \sum_i \phi_i (Re_i) / \sum_i \phi_i' (Re_i') \tag{20}$$

Eq. (20) represents a general formula for converting efficiencies from model to prototype machines. Most of the formulae that have been proposed can be derived from it by appropriate simplifying assumptions. An assumption made by almost all investigators was that a single function of the Reynolds number could be used to represent the friction loss coefficients for both the model and prototype, such that

$$\sum_i \phi_i (Re_i) / \sum_i \phi_i' (Re_i') \equiv \phi (Re) / \phi (Re') \tag{21}$$

then, by Eqs. (20) and (21)

$$\delta/\delta' = (1-\eta_h)/(1-\eta_h') = a + (1-a)\, \phi (Re)/\phi (Re') \tag{22}$$

Eq. (22) may be considered as the starting point for deriving almost any of the simplified formulae for turbines.

D. GENERAL FORMULAE FOR PUMPS

1. Component Losses and Efficiencies

Much of the discussion of hydraulic turbines can be applied to pumps but there are some notable differences. The pump impeller must impart sufficient energy to the fluid to create the pump head, H, and overcome all of the

losses. In addition the impeller must pump the entire leakage flow through essentially the head H. The measurement of the pump head[1] may depend somewhat on the type of installation but it is common practice to charge the losses in a suction elbow or other short intake passage to the pump and to credit the pump with the kinetic energy in the fluid at the discharge flange.

The impeller shrouds usually rotate in contact with the fluid being pumped and this gives rise to the disk friction loss. In the United States it is customary to charge this loss to the hydraulic losses in the pump and account for it by the hydraulic efficiency, η_h. Many British and European authors[2] charge the disk friction to the mechanical losses and account for it by introducing an "internal efficiency", η_i, using the hydraulic efficiency, η_h, to account for the losses associated with the through flow including the recirculation of the leakage flow by the impeller. Thus internal efficiency and hydraulic efficiency may be indentical depending upon the individual author's designation. In this report, the disk friction has been considered to be a hydraulic loss and included in the hydraulic efficiency, η_h.

2. Dimensionless Loss Parameter Ratio

The hydraulic efficiency of a pump is given by

$$\eta_h = H/(H+h) = 1/\left[1 + (h/H)\right] = 1/(1+\delta) \qquad (23)^{3}$$

and the ratio of the dimensionless loss parameters for the prototype and model is given by

$$\delta/\delta' = \left[(1-\eta_h)/(1-\eta_h')\right]\left[\eta_h'/\eta_n\right] = a + (1-a)\,\phi(Re)/\phi(Re') \qquad (24)$$

wherein the same simplifying assumptions have been made as for the derivition of Eq. (22) from Eq. (20). It is seen that the dimensionless loss parameters for either a pump or a turbine may be estimated from the same type of equation but this does not extend to the hydraulic efficiencies. The similarities between pumps and turbines have led many authors to assume that identical functions of the Reynolds number and values of "a", the kinetic part of the

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[1] See, for example "Centrifugal Pumps", ASME Power Test Code PTC 8.1-1954 or "Standards of the Hydraulic Institute, Centrifugal Pump Section," 10th Edition, 2nd Rev.

[2] See, for example, reference [63], pp. 18-21,

[3] See, for example, reference [50], p. 98.

total loss for the model, could be used in Eqs. (22) and (24) so that experience with turbines could be applied directly to pumps. The 1955 Standards of the Hydraulic Institute recommend a turbine-type formula for use with pumps which lends some support to this argument.

E. SIMPLIFICATIONS OF THE GENERAL FORMULAE

1. Condensation of Terms

a. Series Expansion

Many investigators have found that satisfactory results were obtained by setting

$$\phi\ (Re) = \text{Constant}/(Re)^m \tag{25}$$

Experience has indicated that m was a constant over a reasonable range of Reynolds numbers and that the value of m probably depended upon individual practice in constructing and testing machines. The constant in Eq. (25) has been assumed to be the same for geometrically similar machines operated under dynamically similar conditions. Upon substitution of Eq. (25), Eqs. (22) and (24) may take the form

$$\chi = a + (1-a)y^m \tag{26}$$

which may be approximated[1] by,

$$\chi \cong y^{m(1-a)} = y^n \tag{27}$$

by the simple expedient of expanding each equation in an infinite logarithmic series and retaining only the first two terms.

b. Estimate of Error

The error due to neglect of the higher order terms in the development of Eq. (27) cannot be determined by the method given by Pfleiderer[1] because the series converge too slowly. Instead, Fig. 1 shows the error, Δ_1, given by

[1]See, for example, Ref. 9b, p. 628, and Ref. [52], pp. 756-757, discussion by Pfleiderer.

$$\Delta_1 = \left[a + (1-a)y^m\right] - y^n \tag{28}$$

with $n = (1-a)\,m$ and $y = Re'/Re$. The numerical values of a, m, and n used in Fig. 1 are those that have been suggested by several investigators. The ratio $y = Re'/Re$ may be replaced by any of the applicable substitutes, such as D'/D, as will be discussed later under Part E. 2. The hydraulic efficiency of a turbine[1] is given by

$$\eta_h = 1 - (1-\eta_h')\,\delta/\delta' = 1 - (1-\eta_h')\,\chi \tag{29}$$

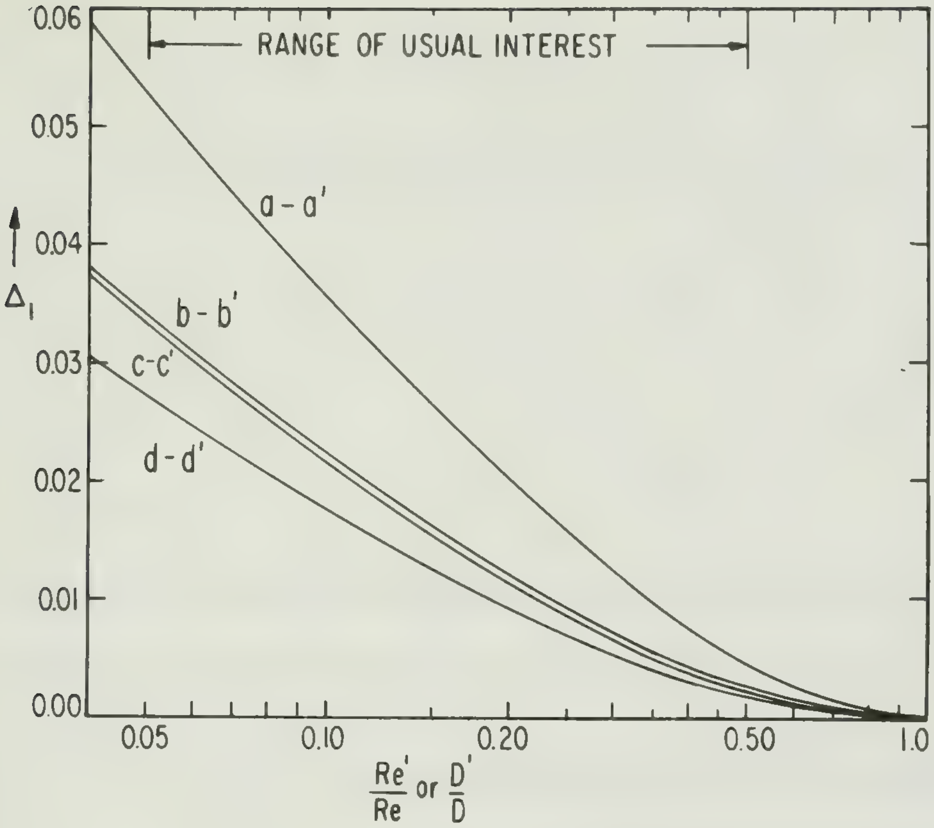
and χ is to be taken from Eq. (27). An estimate of the accuracy of the approximation may be had by letting $a = 1/2$, $m = y = 1/5$, and $n = 1/10$. Several calculated hydraulic efficiencies are shown in Table 1. The constants are those of the Ackeret [35] and Pfeiderer 1947 [34] formulae respectively.[2] Since the numerical values of a, m, and n must ultimately be determined by experience, either of Eqs. (26) or (27) can be adjusted to fit given data with virtually identical accuracy merely by ignoring the condition $n = (1-a)\,m$. As a consequence of the foregoing, many authors have proposed

$$\delta/\delta' = (1-\eta_h)/(1-\eta_h') = (Re'/Re)^n \tag{30}$$

for turbines. The corresponding equation for pumps is

$$\delta/\delta' = \left[(1-\eta_h)/(1-\eta_h')\right]\left[\eta_h'/\eta_h\right] = (Re'/Re)^n \tag{31}$$

[1] The argument may be extended to pumps by an appropriate change in Eq. (29)
[2] See also Tables 2 and 3.



$$\Delta_1 = \left[a + (1-a)\, y^m\right] - y^n$$

$$y = Re'/Re \text{ or } y = D'/D$$

$$n = (1-a)\, m$$

CURVE LETTER		
a.	$\delta/\delta' \cong (1-\eta)/(1-\eta') = 0.25 + 0.75\,(Re'/Re)^{0.25}$	
a'.	$\delta/\delta' \cong (1-\eta)/(1-\eta') = (Re'/Re)^{3/16}$	
b.	$\delta/\delta' \cong (1-\eta)/(1-\eta') = 0.25 + 0.75\,(D'/D)^{1/3}$	MOODY
b'.	$\delta/\delta' \cong (1-\eta)/(1-\eta') = (D'/D)^{0.25}$	MOODY (1925)
c.	$\delta/\delta' = 0.5 + 0.5\,(Re'/Re)^{0.20}$	ACKERET
c'.	$\delta/\delta' = (Re'/Re)^{0.10}$	PFLEIDERER (1947)
d.	$\delta/\delta' = 0.3 + 0.7\,(Re'/Re)^{0.20}$	HUTTON
d'.	$\delta/\delta' = (Re'/Re)^{0.14}$	PFLEIDERER (1954)

FIG. 1. ERROR Δ_1 DEFINED BY EQ. (28)

TABLE 1

CALCULATED HYDRAULIC EFFICIENCIES

Re'/Re	1/5			1/10			1/20		
η_h'	0.900	0.800	0.700	0.900	0.800	0.700	0.900	0.800	0.700
η_h by Eqs. (27) and (29)	0.915	0.830	0.745	0.921	0.841	0.762	0.926	0.852	0.778
η_h by Eqs. (26) and (29)	0.914	0.828	0.741	0.918	0.837	0.755	0.923	0.845	0.768

2. Substitutions for Reynolds Number

a. Constant Kinematic Viscosity

In cases where cold water was the fluid used in both the model and prototype, changes in the kinematic viscosity usually have had very little effect on the hydraulic efficiency. This may not have been true if one wheel diameter was quite small[1] and temperature changes were quite large. Under the assumption that $\nu' = \nu$, Eqs. (30) and (31) may be written

$$\delta/\delta' = (1-\eta_h)/(1-\eta_h') = (D'/D)^n (H'/H)^{n/2} \qquad (32)$$

for turbines, and, for pumps,

$$\delta/\delta' = \left[(1-\eta_h)/(1-\eta_h')\right]\left[\eta_h'/\eta_h\right] = (D'/D)^n (H'/H)^{n/2} \qquad (33)$$

[1] Riemerschmid [14, 16] reported three percent increase in the efficiency of a small Francis turbine, D_{throat} = 4.3 inches, n_s = 75 at best efficiency, for an increase in the Reynolds number of 240 percent due to changing the water temperature. Ippen [32], page 829, Fig. 8, found about one percent increase in the efficiency of a small pump, D = 8 inches, D_o = 5-3/16 inches, n_s = 1991 at best efficiency, for an increase in the Reynolds number of 50 percent obtained by increasing the wheel speed.

b. Constant Kinematic Viscosity and Constant Head.

In addition to the case of nearly constant fluid viscosity mentioned above, the speed of the model sometimes has been chosen so that both model and prototype heads were very nearly equal. In such cases, Eqs. (32) and (33) may be written

$$\delta/\delta' = (1-\eta_h)/(1-\eta_h') = (D'/D)^n \tag{34}$$

for turbines and, for pumps,

$$\delta/\delta' = \left[(1-\eta_h)/(1-\eta_h')\right]\left[\eta_h'/\eta_h\right] = (D'/D)^n \tag{35}$$

c. Hydraulic Losses Independent of Reynolds Number

It has been common practice to try to approximate the hydraulic losses in machines by analogy to pipe flow or boundary layer flow over flat plates. Pipes or flat plates that are hydraulically rough exhibit constant friction coefficients at sufficiently large Reynolds numbers. The Reynolds number for a pipe is usually defined by

$$Re = V d/\nu \tag{36}$$

wherein,

V = average velocity over the cross-sectional area,

d = diameter, and

ν = kinematic viscosity of the fluid.

The Reynolds number for a flat plate is given by

$$Re = V x/\nu \tag{37}$$

wherein,

V = Free stream velocity (outside the boundary layer),

x = distance from leading edge of plate measured in the direction of flow, and

ν = kinematic viscosity of the fluid.

Unlike a pipe for which the Reynolds number is constant along the length, the Reynolds number for a plate increases from zero at the leading edge to a maximum at $x = \ell$, the length of the plate parallel to the direction of flow.

The friction coefficients for rough pipes and plates become independent of Reynolds number at sufficiently large values of that parameter even though the initial turbulence may be quite low. This has led many investigators to the conclusion that friction coefficients for turbines and pumps should be independent of Reynolds number because of high initial turbulence in the flow, inevitable sudden changes in direction or cross-sectional area of the flow passages, and, particularly in older machines, considerable surface roughness. Also in support of this view is the fact that losses in vane cascades tend to become independent of Reynolds number above Re $\cong$ 600, 000. The Reynolds number of a cascade is given by Eq. (37) with $x = \ell$ = the chord of the vane profile and V = the velocity of the fluid just ahead of the vanes. If the fluid is cold water, V = 25 ft/sec and ℓ = 3 inches should be sufficient to render the losses independent of the Reynolds number. These values are entirely reasonable for wheels of 12 inches or larger diameter. Thus, if the cascade analogy holds for turbines and pumps, the losses in the blading should be nearly independent of Reynolds number in most cases of practical interest. This would appear to indicate that any Reynolds number effects associated with most pumps or turbines came from the vane-free sections of the machines.

d. Friction Losses Partly Dependent on Reynolds Number

It should be noted that all the losses in some parts of a machine might be independent of Reynolds number while the losses in other parts could be quite dependent on the Reynolds number. In fact, the wide variety of possible sizes of machines, materials, surface finishes, average flow velocities, etc., indicates that the dependence of friction losses on Reynolds number can be anything from total to negligible. If the dependence on Reynolds number is the same for a model and its prototype, Eqs. (22) and (24) for a turbine and a pump respectively, should apply with $\phi(\mathrm{Re}) = \mathrm{constant}/(\mathrm{Re})^m$, or Eqs. (34) and (35) could also be used with good accuracy provided that the conditions for use of the latter equations are fulfilled.

3. Other Simplifications

a. Assumption that the ratio $\eta_h'/\eta_h \cong$ unity.

The majority of authors have assumed that the ratio η_h'/η_h could be set equal to unity in all of the equations for pumps. For example, Eq. (35)

would reduce to Eq. (34) so that pumps and turbines would be treated in the same way. Medici [31] and Pantell [50] appear to be the only investigators who have questioned this heretofore. Fig. 2 shows the error, Δ_2, introduced by setting η'_h/η_h equal to unity in any of the equations for pumps, i.e. using an unaltered turbine-type formula to predict pump efficiency. Admittedly, the errors are small for the usual range of interest but, since they are easily eliminated, there appears to be no valid argument for using an equation of the form of Eq. (34) with pumps.

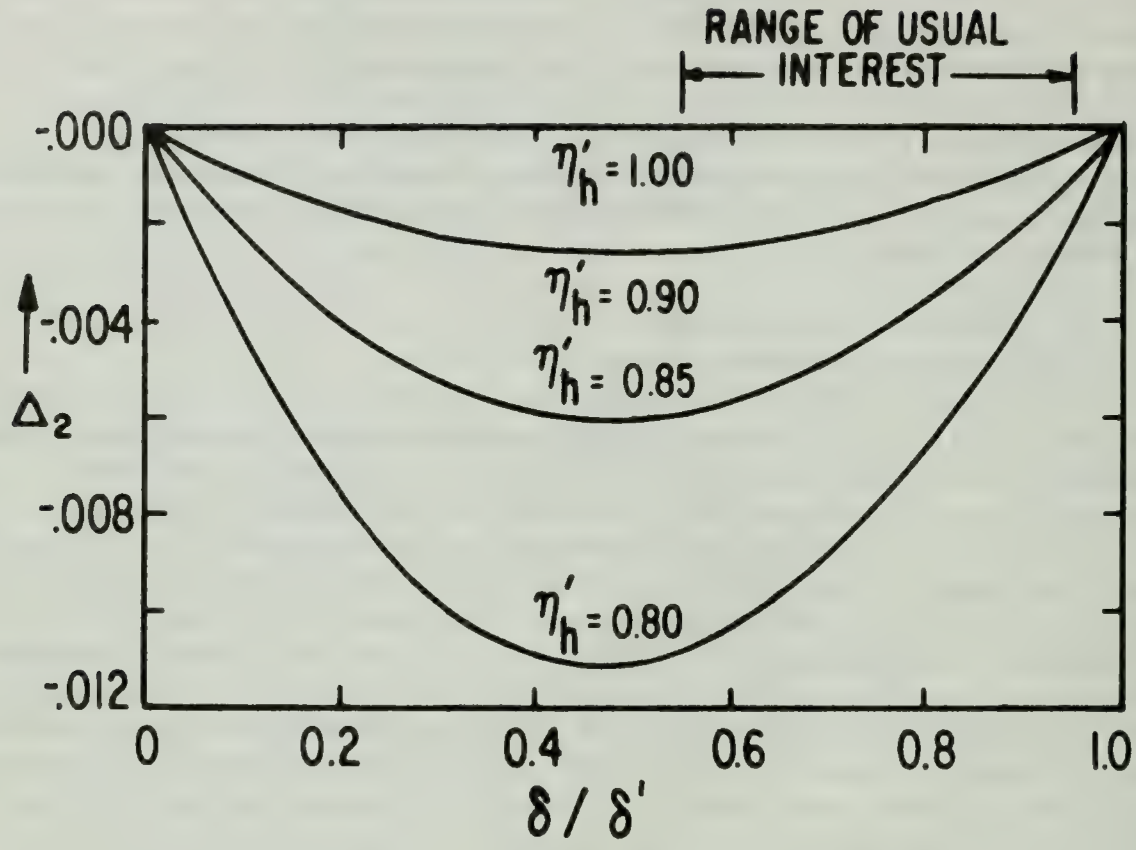
b. Assumption that $\eta_h \cong \eta$.

Moody [9, 17, 23] in particular, among several authors, assumed $\eta_m \cong \eta_v \cong$ unity so that $\eta_h \cong \eta$ and applied the conversion formula to the over-all efficiency rather than to the hydraulic efficiency. This expedient was bred of practical necessity because of the difficulty in estimating η_v and η_m for either the model or prototype. Very few authors have given any numerical values for η_v or η_m. Staufer [8] mentioned the value $\eta'_m = \eta_m = 0.98$. Gregorig [16] quoted $\eta_m = 0.93$ for the turbine tested by Riemerschmid (D_{throat} = 4.3 inches) and $\eta_m = 0.98$ for two other turbines having diameters of 20 inches and 134 inches respectively. Mühlemann [35] reported η_m = 0.994 for a model turbine of unspecified size. He stated that the mechanical losses in a large prototype turbine should only be 0.2% to 0.3% whereas those for the model might be from 0.5% to 1.5% or more depending upon the construction details. The Society of German Engineers[1] (VDI) states that $0.98 \geqq \eta_m \geqq 0.90$ may be used depending on the size of the pump. No values of η_v were found in the literature consulted. Some authors appeared to lump η_m and η_v together in η_m but with no indication of even approximate values for either model or prototype.

F. CONVERSION FORMULAE FOR TURBINES AND PUMPS.

Table 2 contains a listing of formulae taken from the literature. Wherever possible the original reference has been consulted but, in a few cases it has been necessary to rely on secondary sources such as the excellent paper by Hutton [52]. Unless otherwise noted, the formulae were proposed for use with hydraulic turbines but many of these have been used with pumps, apparently with adequate success [60, 65]. A graphical comparison of several of the formulae listed in Table 2 is shown in Fig. 3. Table 3 lists prototype efficiencies computed by several formulae from the same assumed model data. A discussion of the individual formulae as well as consideration of more detailed methods of loss separation will be found in Appendix II (Section H.)

[1] Page 20 of reference [47].



$$\Delta_2 = \frac{1}{1 + \left[(1/\eta'_h)-1\right] (\delta/\delta')} - \left[1 - (1-\eta'_h)(\delta/\delta')\right]$$

FIG. 2. ERROR Δ_2 DUE TO SETTING $\eta'_h/\eta_h = 1$ IN EQ. (24)

TABLE 2

EFFICIENCY CONVERSION FORMULAE

Formulae for the Point of Maximum Efficiency

NAME	DATE	REF.[1]	FORMULA
Camerer	1909	2, 3, 52	$\frac{\delta}{\delta'} = \frac{0.12 + F/\sqrt{m} + 2.5\nu/(100F+2)v\sqrt{m}}{0.12 + F'/\sqrt{m'} + 2.5\nu'/(100F'+2)v'\sqrt{m'}}$
			F = friction factor in Biel Formula $\cong 0.015\ (\text{meters})^{1/2}$
			m = hydraulic radius[2] in meters
			v = average velocity, meters/second
			ν = kinematic viscosity of fluid, square meters/second
		9a, 9b	The third term in both numerator and denominator was found to be small and usually omitted.
	1924	52	$\frac{\delta}{\delta'} = \frac{0.12 + F/\sqrt{m}}{0.12 + F'/\sqrt{m'}}$
			with F = 0.27, this formula was attributed to Biel by Rütschi [41]
	1927	10, 12	$\delta/\delta = \left[1.4 + (1/\sqrt{D})\right] / \left[1.4 + (1/\sqrt{D'})\right]$
			D = Runner diameter in meters
Staufer	1925	8, 9b	$\delta/\delta' = \left[(D'/D)\sqrt{H'/H}\right]^{0.25}$

[1] Footnotes are given on Page 28.

TABLE 2 (Continued)

NAME	DATE	REF. [1]	FORMULA
Moody[3]	1925	9a, 9b	$(1-\eta)/(1-\eta') = 0.25 + 0.75\,(D'/D)^{1/3}$
		9, 12, 15, 24	$(1-\eta)/(1-\eta') = (D'/D)^{0.25}\,(H'/H)^{0.10}$
		12, 25, 38	$(1-\eta)/(1-\eta') = (D'/D)^{0.25}\,(H'/H)^{0.01}$
		12, 17, 25, 33, 38, 44	$(1-\eta)/(1-\eta') = (D'/D)^{0.25}$
	1935	17, 29	$(1-\eta)/(1-\eta') = \left[(\log 800D')/(\log 800D)\right]^2$
	1942	24, 25, 29, 43, 70	$(1-\eta)/(1-\eta') = (D'/D)^{0.20}$
Haeger	1927	10	$\delta/\delta' = (kD'/k'D)^{0.314}\left[(\nu/\nu')(H'/H)^{0.125}\right]^q$ $q \cong 0.314$; k = a measure of the surface roughness
Oesterlin	1928	11	$\delta/\delta' = (D'/D)^{0.314}$ Haeger's formula for case where flow through both turbines is in the complete turbulence zone.
Ackeret	Circa 1930	35, 38	$\delta/\delta' = 0.5 + 0.5\,(Re'/Re)^{0.20}$ Published by Mühlemann
Shōgenji	1931	46	$\frac{1-\eta}{1-\eta'} = \left(\frac{D'}{D}\right)^{0.23}\left(\frac{H'}{H}\right)^{0.115}$

[1] Footnotes are given on Page 28.

TABLE 2 (Continued)

NAME	DATE	REF.[1]	FORMULA
Pfleiderer[4]	1932	15	$(1-\eta)/(1-\eta') = (D'/D)^{0.45}\ (n'/n)^{0.20}$
	1947	34, 52	$\delta/\delta' = (Re'/Re)^{0.10}$ Approximation of Ackeret's formula.
	1949	41	$\delta/\delta' = (Re'/Re)^{0.10}\ (D'/D)^{0.05}$
	1954	52	$\delta/\delta^1 = (Re'/Re)^{0.14}$ Approximation of Hutton's formula.
Gregorig	1933	16	$\left[\delta - (V_3^2/2gH)\right] / \left[\delta' - (V_3'^2/2gH')\right] = (Re'/Re)^{0.25}$ V_3 = Average velocity at exit from draft tube
Pardoe	1941	24	$(1-\eta)/(1-\eta') = (D'/D)^{0.20}$
		27	$(1-\eta)/(1-\eta') = (D'/D)^{0.25}\ (H'/H)^{0.125}$ $(1-\eta)/(1-\eta') = (D'/D)^{0.20}(H'/H)^{0.01}$
Komoriya and Hatori	1942	46	$\eta = \eta' \left[1 + C\ (1-\eta')\ \log\ (D/D')\right]$ C = 0.7 for Francis turbines C = 0.8 for propeller turbines
Medici	1943	31, 52	$\delta/\delta' = (D'/D)^{0.25}\ (H'/H)^{0.10}$ for turbines
		31, 50	$\eta = \dfrac{1}{1 + \left[(1/\eta') - 1\right]\ (D'/D)^{0.25}\ (H'/H)^{0.10}}$ for pumps[5]
	1949	38	$Q = Q'\ (n/n')(D/D')^3\ f\left[(D/D')^2\ (n/n')\right]$ for turbines Q = Discharge; n = speed; f = function shown in Fig. 4 .

[1] Footnotes are given on Page 28.

TABLE 2 (Continued)

NAME	DATE	REF.[1]	FORMULA
Canaan	1945	52	$\delta/\delta' = 0.5 + 0.5\,(Re'/Re)^{0.25}$
Nechleba	1949	52	$\frac{\delta}{\delta'} = \frac{1 + 52\,(\epsilon/D)^{0.50}}{1 + 52\,(\epsilon'/D')^{0.50}}$ ϵ = a linear measure of the surface roughness
McDonald	1950	40, 52	$\frac{\delta}{\delta'} = \frac{1 + c/(\log Re)^{2.58}}{1 + c/(\log Re')^{2.58}}$ All surfaces smooth[6] $\frac{\delta}{\delta'} = \frac{1 + c/\left[\log (fD/\epsilon)\right]^2}{1 + c/\left[\log (fD'/\epsilon)\right]^2}$ All surfaces rough[6, 7]
Rütschi	1951	18, 41, 50	$\frac{\eta_h}{\eta_h'} = \frac{f}{f'} \cong \frac{1 + \left[3.15/(D_o')^{1.6}\right]}{1 + \left[3.15/(D_o)^{1.6}\right]}$ or use Fig. 5
	1951	45	$\frac{\eta_h}{\eta_h'} = \frac{f}{f'} \cong \frac{1 - \left[3.15/(D_o)^{1.6}\right]}{1 - \left[3.15/(D_o')^{1.6}\right]}$ or use Fig. 5
Rütschi & Pfleiderer	1951	45, 63	$\eta_h = 1 - \left[1 - \eta_h'\,(f/f')\right]\,(Re'/Re)^n$ $\frac{f}{f'} \cong \frac{1 - \left[2.21/(D_o)^{1.5}\right]}{1 - \left[2.21/(D_o')^{1.5}\right]}$ or use Fig. 5 $n \cong 0.1$ or use Fig. 13 D_o = Eye or inlet diameter of impeller in centimeters

[1] Footnotes are given on Page 28.

TABLE 2 (Continued)

NAME	DATE	REF.[1]	FORMULA
German Society of Engineers Test Code for Centrifugal Pumps	1952	47	$\eta_h = \eta/\eta_m = 1 - (1-\eta_h') \left[(\eta'/\eta)(\nu/\nu')\right]^{0.1}$ For a given pump tested in the range $1/15 < \left\|(\eta'/\eta)(\nu/\nu') = (Re'/Re)\right\| < 15$ $\eta_m = 0.90$ to 0.98 depending on size of pump. $\eta_h = 1 - (1-\phi\eta_h') \left[(N'/N)(D'/D)\right]^{0.1}$ $\phi = \dfrac{1 - \left[70/(D_o)^{1.5}\right]}{1 - \left[70/(D_o')^{1.5}\right]}$ D_o = Eye or inlet diameter of impeller in millimeters. The model and prototype must be tested with the same fluid. The affinity laws $Q/ND^3 = Q'/N'D'^3$ and $H/(ND)^2 = H'/(N'D')^2$ may be assumed valid over the range $1/15 < (Re/Re') < 15$
Pantell[8]	1953	50	$\eta = \dfrac{\eta_m}{1 + \left[(\eta_m'/\eta') - 1\right] \left[Re'/Re\right]^{0.25}}$ For smooth surfaces[9]. $\eta = \dfrac{\eta_m}{1 + \left[\eta_m'/\eta' - 1\right]\left[(k_F/k_F')(D'/D)\right]^{0.314}}$ For rough surfaces[10] Values of k_F and k_F' are shown in Fig. 6

[1] Footnotes are given on Page 28.

TABLE 2 (Continued)

NAME	DATE	REF.[1]	FORMULA
Byron Jackson Pumps, Inc.[11]			$(1-\eta)/(1-\eta') = (D'/D)^{0.165}$
Sulzer Brothers[11]			$(1-\eta)/(1-\eta') = (D'/D)^{1/6.5}(H'/H)^{1/28}$ $= \left[(D'/D)\left(\sqrt[4.3]{H'/H}\right)\right]^{1/6.5}$
Formulae for Predicting Overall Performance			
Miyagi	1937	19, 42, 46, 48, 49	$\eta = \eta' + (1-\eta'_o)\left(\frac{Q'}{Q'_o}\right)^p \left[1 - \left(\frac{D'}{D}\right)^{0.25}\left(\frac{H'}{H}\right)^{0.125}\right]$ p = 1.75 for Francis Turbines P = 0.75 for Propeller Turbines p = -0.125 for Kaplan Turbines Subscript o refers to maximum efficiency point of model turbine
Fuchizawa	1951	42	$\eta_h = \eta'_h + (1-\eta'_{h_o})\left[1 - (\frac{\epsilon}{\epsilon'})^{0.25}(\frac{D'}{D})^{0.25}\right]$ Usually $\epsilon = \epsilon'$ Subscript o same as for Miyagi's formula
Yamazaki	1952	46, 48	$\frac{1-\eta}{1-\eta'} = \chi\left[1 + \frac{(1-\chi)(Q_o-Q)}{\chi Q_o}\right]$ First Formula $\frac{1-\eta}{1-\eta'} = \chi\left[1 + \frac{(1-\chi)(Q_o-Q)^2}{\chi Q_o^2}\right]$ Second Formula $\chi = (D'/D)^{0.2}(H'/H)^{0.1}$ Subscript o same as for Miyagi's formula

[1] Footnotes are given on Page 28.

TABLE 2 (Continued)

NAME	DATE	REF.[1]	FORMULA
Hirotsu	1952	48, 49	$\eta/2 = -q^x\zeta + \sqrt{2q^x\zeta - X\left[(1-\eta') - (1-q^x\zeta' - \eta'/2)^2\right]}$ $q = Q/Q_o$ $x = 1$ for Francis and propeller turbines $x = 1/2$ for Kaplan turbines $X = X_f = (D'/D)^{0.25}\,(H'/H)^{0.125}$ for Francis turbines $X = X_p = (D'/D)^{0.20}\,(H'/H)^{0.10}$ for propeller and Kaplan turbines $\zeta = 1 - (\eta_o'/2)$ for Francis turbines $\zeta = \left[1 + X_p\,(1-\eta_o')\right]/2$ for propeller and Kaplan turbines $\zeta' = 1 - (\eta_o'/2)$ Subscript o refers to maximum efficiency point of model turbine.
Hutton	1954	52, 70	$\delta/\delta' = a + b\,(Re'/Re)^{0.20}$ Recommended for Kaplan and propeller-type turbines. a = 0.3 and b = 0.7 at the maximum efficiency point. Values of a and b are given in Fig. 20 for other operating conditions.
Itaya, Tejima, and Nishikawa	1961	61	$\eta_h = \eta_{h1}' + (\eta_{h2}' - \eta_{h1}')\dfrac{1 - (Re_1'/Re)^{1/n}}{1 - (Re_1'/Re_2')^{1/n}}$ Probably n = 4 for smooth surfaces. Subscripts 1 and 2 refer either to the same model operated at different speeds N_1', and N_2' (preferably $N_2' \geqq 2N_1'$) or to two different models having diameters D_1' and D_2' respectively.

[1] Footnotes are given on Page 28.

[1]Numbers refer to the bibliography in Appendix I (Section G.)

[2]See Ref. 9b p. 625

[3]It is difficult to assess correct dates because it appears that Moody used some of the formulae for several years before publishing them.

[4]Pfleiderer's 1932 formula was obtained by setting $H'/H = (ND')^2/(ND)^2$ in the third Moody formula

[5]This appears to be the first published formula specifically for pumps

[6]Values of c, f, and ϵ, the measure of the surface roughness must be determined by experiment. $c = (1-a)/a$, where a is defined by Eq. (22), is a measure of the surface friction losses.

[7]If unity is neglected in comparison with the second term in both numerator and denominator and $f/\epsilon = 800$, the formula reduces to the Moody 1935 formula.

[8]Pantell stated that he had developed a formula of this type in 1942 during a series of lectures sponsored by the German Society of Engineers (V.D.I.)

[9]Based on a pipe friction formula due to H. Blasius, Ref. [4], page 12

[10]Based on a pipe friction formula due to K. Fromm, Ref. [6], pp. 352, 353

[11]Unpublished formula

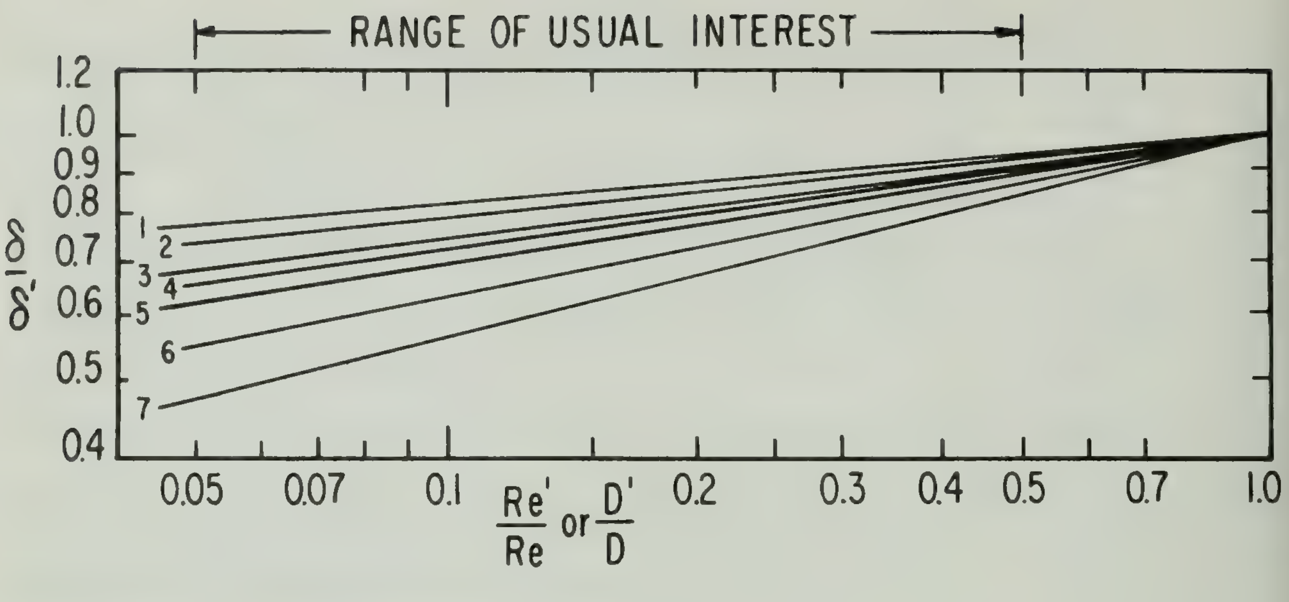
TABLE 3

TURBINE AND PUMP EFFICIENCIES PREDICTED BY TYPICAL FORMULAE

D'/D or Re'/Re as required by Formula			0.20			0.10			0.05		
η' or η'_h as required by Formula			0.900	0.800	0.700	0.900	0.800	0.700	0.900	0.800	0.700
NAME	DATE	FORMULA WITH H'=H IF NECESSARY									
Moody	1925	$(1-\eta)/(1-\eta') = (D'/D)^{0.25}$	0.933	0.866	0.799	0.944	0.888	0.831	0.953	0.905	0.858
Staufer	1925	$(1-\eta_h)/(1-\eta'_h) = (D'/D)^{0.25}$	0.933	0.866	0.799	0.944	0.888	0.831	0.953	0.905	0.858
Ackeret	Circa 1930	$(1-\eta_h)/(1-\eta'_h) = 0.5 + 0.5\,(Re'/Re)^{0.20}$	0.914	0.828	0.741	0.918	0.837	0.755	0.923	0.845	0.768
Pfleiderer	1932	$(1-\eta_h)/(1-\eta'_h) = (Re'/Re)^{0.10}$	0.915	0.830	0.745	0.921	0.841	0.762	0.926	0.852	0.778
Moody	1942	$(1-\eta)/(1-\eta') = (D'/D)^{0.20}$	0.928	0.855	0.783	0.937	0.874	0.811	0.945	0.890	0.835
Canaan	1945	$(1-\eta_h)/(1-\eta'_h) = 0.5 + 0.5\,(Re'/Re)^{0.25}$	0.917	0.833	0.750	0.922	0.844	0.766	0.926	0.853	0.779
Hutton	1954	$(1-\eta_h)/(1-\eta'_h) = 0.3 + 0.7\,(Re'/Re)^{0.20}$	0.919	0.839	0.758	0.926	0.852	0.777	0.932	0.863	0.795
Pfleiderer	1954	$(1-\eta_h)/(1-\eta'_h) = (Re'/Re)^{0.14}$	0.920	0.840	0.761	0.928	0.855	0.783	0.934	0.869	0.803
		FORMULAE FOR PUMPS ONLY									
Medici[1]	1943	$\eta = \dfrac{1}{1 + \left[(1/\eta') - 1\right] (D'/D)^{0.25}}$	0.931	0.856	0.777	0.941	0.877	0.806	0.950	0.894	0.831
Byron Jackson		$(1-\eta)/(1-\eta') = (D'/D)^{0.165}$	0.923	0.847	0.800	0.931	0.863	0.795	0.939	0.878	0.817
Sulzer Bros.		$(1-\eta)/(1-\eta') = (D'/D)^{1/6.5}$	0.922	0.844	0.766	0.930	0.860	0.789	0.937	0.874	0.811
Eq. (4)		$\eta_h = \dfrac{1}{1 + \left[(1/\eta'_h) - 1\right] (D'/D)^{0.20}}$	0.925	0.847	0.763	0.934	0.864	0.787	0.942	0.879	0.809
Eq. (5)		$\eta = \dfrac{0.995}{1 + \left[(1/\eta'_h) - 1\right] (D'/D)^{0.20}}$	0.921	0.842	0.759	0.930	0.859	0.783	0.938	0.875	0.805

[1] This is the same as Pantell's formula for smooth surfaces with $\eta'_m = \eta_m = 1.000$.

8-29



$$\delta/\delta' = (1-\eta_h)/(1-\eta_h') \text{ FOR TURBINES}$$

$$\delta/\delta' = (1-\eta_h)/(1-\eta_h')\,(\eta_h'/\eta_h) \text{ FOR PUMPS}$$

CURVE NO.

1.	$\delta/\delta' = 0.5 + 0.5\,(Re'/Re)^{0.20}$	ACKERET
2.	$\delta/\delta' = (Re'/Re)^{0.10}$	PFLEIDERER (1947)
3.	$\delta/\delta' = 0.3 + 0.7\,(Re'/Re)^{0.20}$	HUTTON
4.	$\delta/\delta' = (Re'/Re)^{0.14}$	PFLEIDERER (1954)
5.	AVERAGE OF MOODY (1925) AND ACKERET FORMULAE WITH $H = H'$ AND $\nu = \nu'$	
6.	$\delta/\delta' \cong (1-\eta)/(1-\eta') = (D'/D)^{0.20}$	MOODY (1942)
7.	$\delta/\delta' \cong (1-\eta)/(1-\eta') = (D'/D)^{0.25}$	MOODY (1925) WITH $H' = H$
	$\delta/\delta' = (D'/D)^{0.25}$	STAUFER WITH $H' = H$

PRIMED QUANTITIES REFER TO MODEL

$Re = D\sqrt{H}/\nu$

FIG. 3. COMPARISON OF EFFICIENCY CONVERSION FORMULAE ($H/H'=1$)

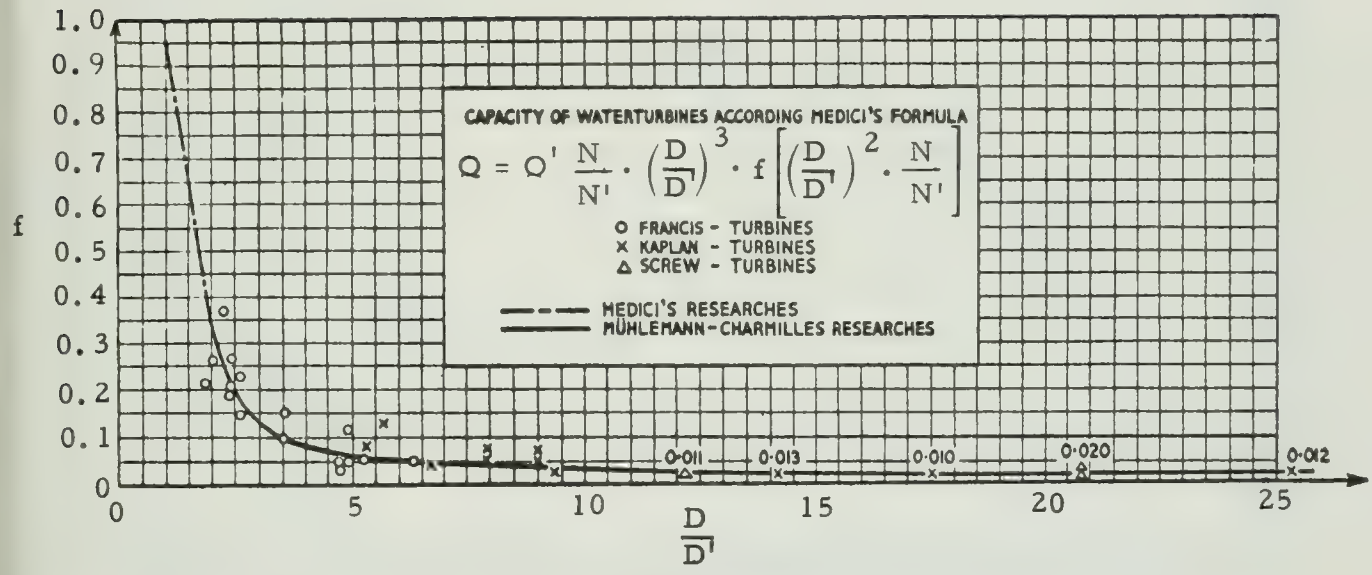


FIG. 4. f-FUNCTION FOR MEDICI FORMULA [38]

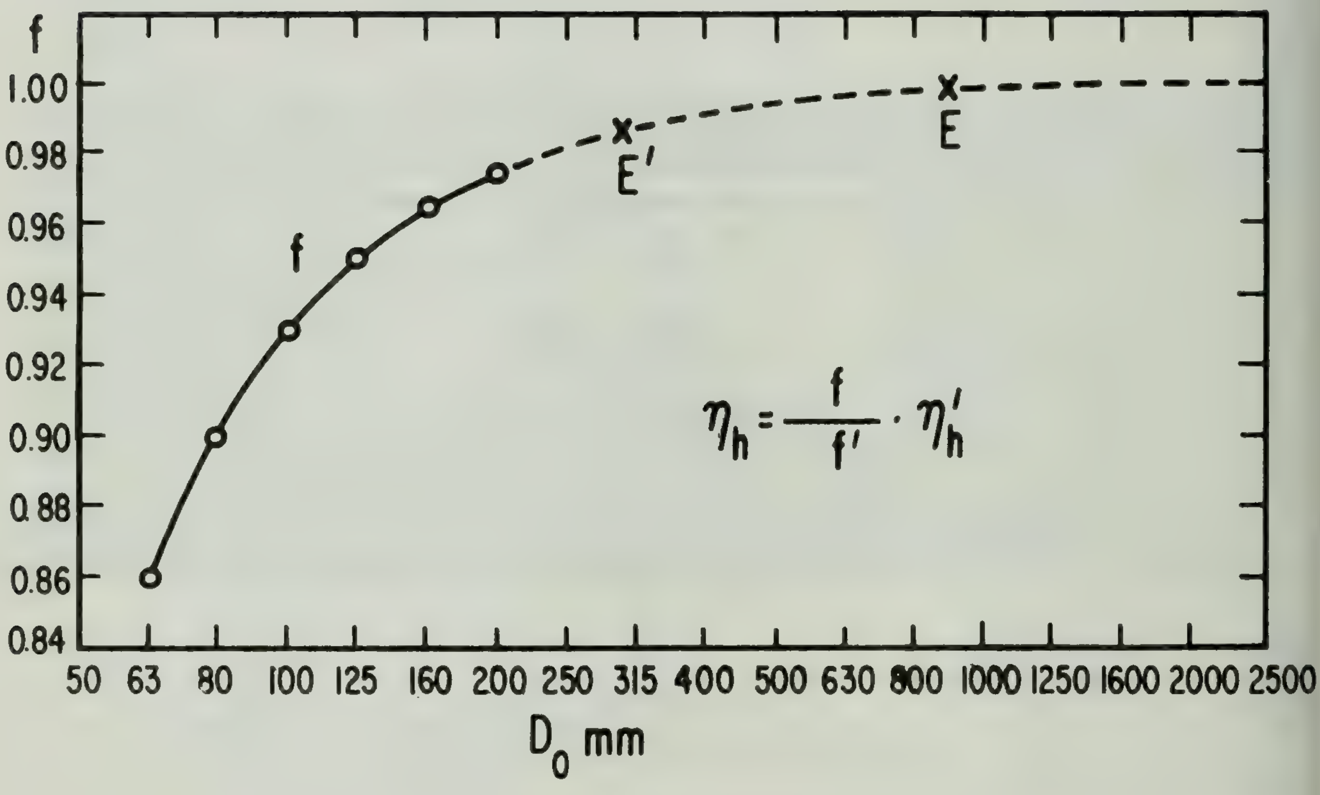
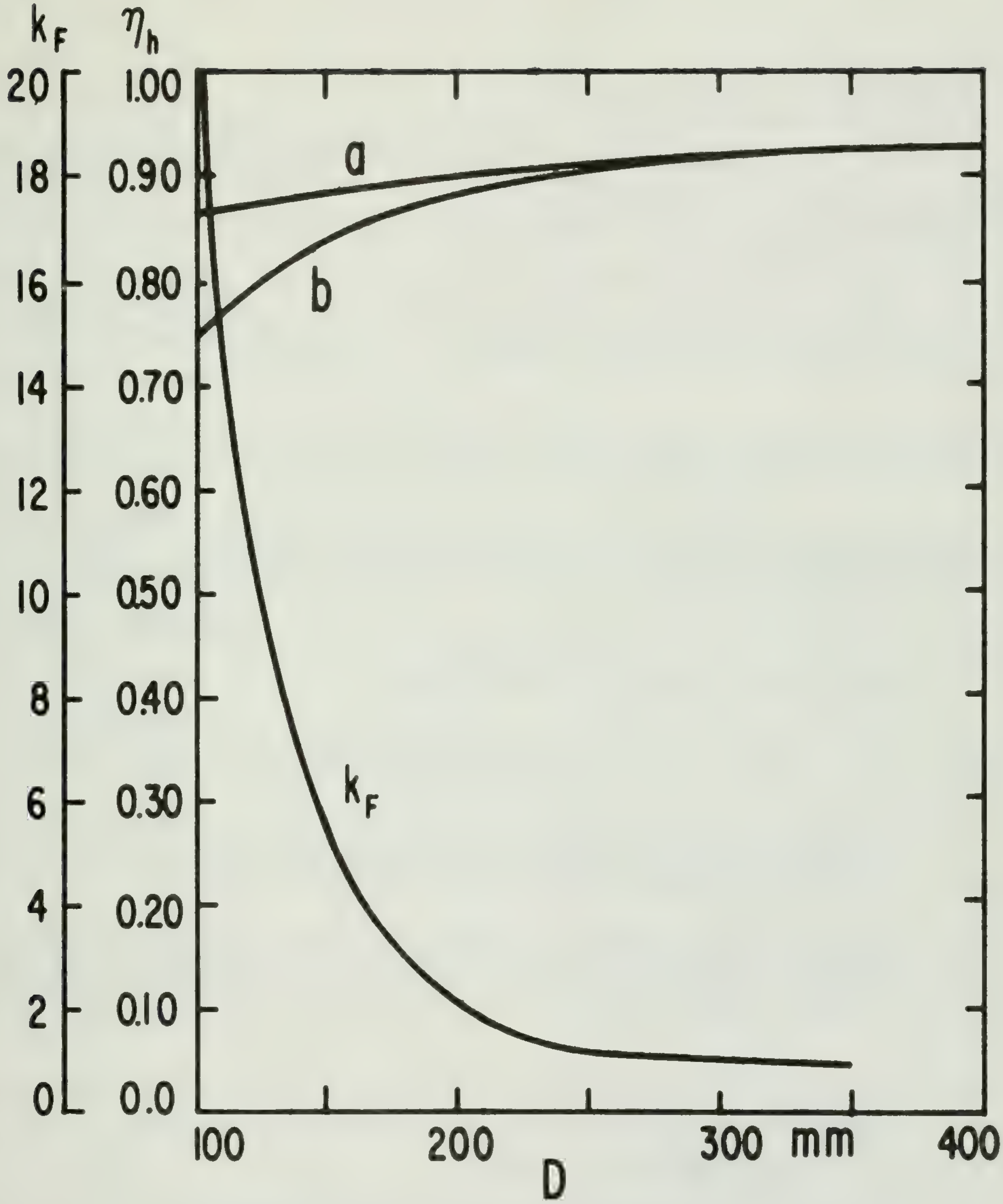


FIG. 5. f-FUNCTION FOR RÜTSCHI FORMULA [41]



CURVE a. η_h FOR SMOOTH SURFACES COMPUTED BY EQ. (39)
CURVE b. η_h FROM RÜTSCHI [41]. SEE FIG. 10, η_s = 5160
k_F COMPUTED BY EQ. (40) WITH VALUES OF η_h FROM CURVE b.

FIG. 6. VALUES OF η_h AND k_F FOR SMALL PUMPS FROM PANTELL [50]

G. APPENDIX I -- BIBLIOGRAPHY

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H. APPENDIX II -- DISCUSSION OF LITERATURE

1. Formulae for the Point of Maximum Efficiency

Any study of efficiency conversion or scale effect is, in some measure, a study of the separation and evaluation of the losses in pumps and turbines. Very likely the difficulties encountered in a detailed analysis led most investigators to quite simplified approaches to the problem. Camerer [2, 3] appears to have considered only the surface friction losses and based their evaluation on a somewhat complicated formula for pipe friction due to Biel [1]. Staufer [8] introduced separate mechanical efficiencies for the model and prototype. He evaluated the surface friction loss by a formula due to Blasius [4] which applied only to smooth pipes for Reynolds numbers ($Re = Vd/\nu$) not exceeding 100,000. Moody [9] may have been the first to have separated the losses into a kinetic part and a surface friction part. He elected to combine these as described in E.1. and also found that the exponent m of the ratio $(H'/H)^m$ became small and difficult to determine precisely from the data available to him. Accordingly, m was set equal to zero and the exponent n of the ratio $(D'/D)^n$ was made 0.25. Moody estimated the surface friction loss by pipe friction formulae proposed by Forcheimer and by Strickler but the value of n = 0.25 was fixed by actual tests on model and prototype turbines.

Moody's approach to scale effect assumed that the model would be large enough to insure a level of turbulence comparable to the complete turbulence zone for rough pipes. Apparently the 16-inch models and laboratory test procedures of the I. P. Morris Company fulfilled these conditions as shown by Table 4 and Fig. 7. Moody's 1935 formula was based on the Karman-Prandtl formula for complete turbulence in rough pipes[1] but the results differed so little from those given by the 1925 formula that Moody did not recommend it.

It was customary to use the difference between the head-water and tail-water elevations as the turbine head for a model test but to credit the prototype turbine with the velocity head at the draft tube exit. This practice, and the apparent tendency of the model efficiencies as determined in the I.P. Morris laboratory to be somewhat low, probably accounted for the fact that Moody's formulae have been criticized for overestimating the step-up between model and prototype efficiencies.[2] In 1941, both Pardoe [24, 27] and Moody [25] recommended n = 0.20 which decreased the step-up (compare curves 6 and 7 of

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[1]Eq. (3), page 139, of Reference [21] . See also Reference [55]
[2]Pages 729 and 730 of Reference [17] , discussion by S. L. Kerr.

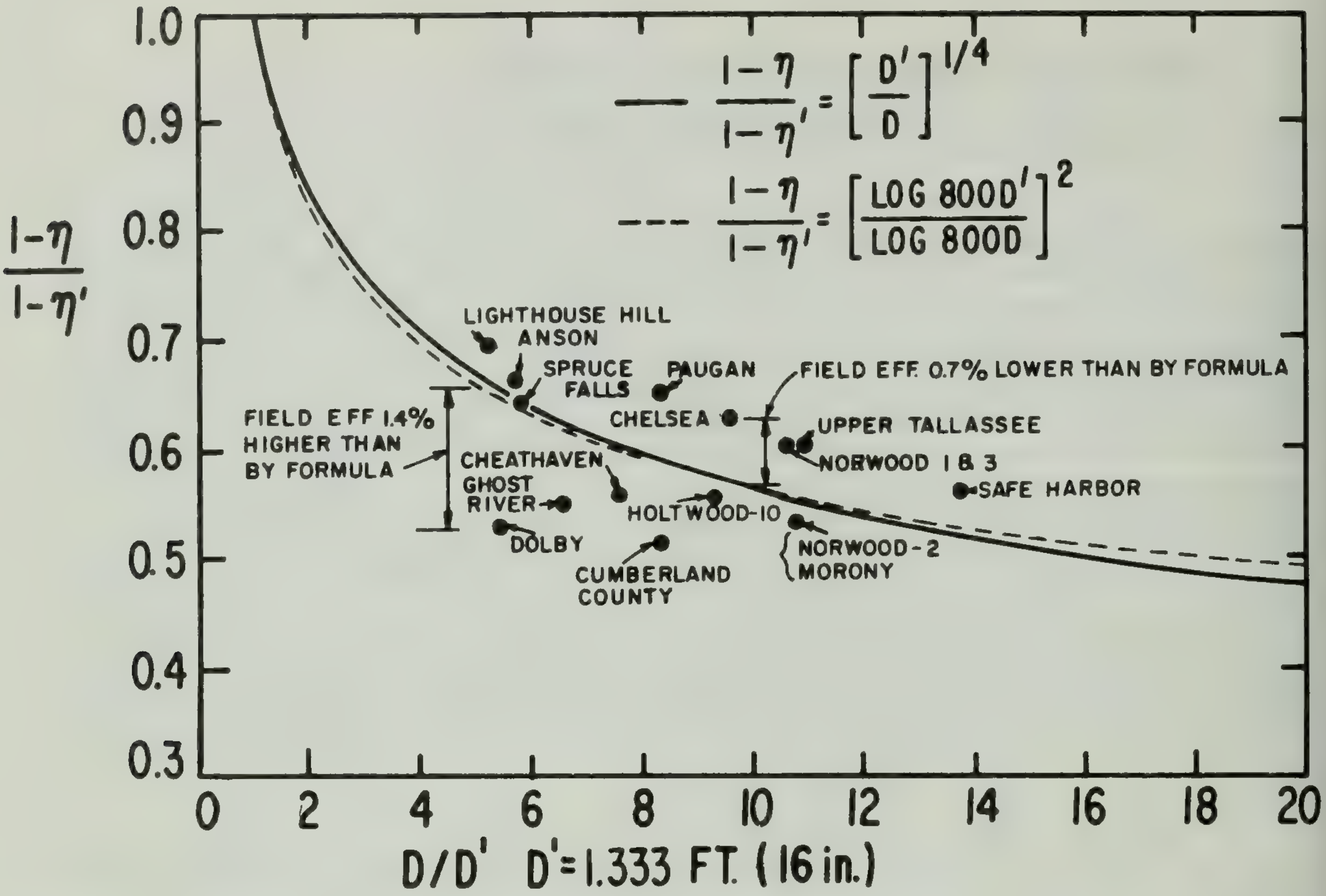


FIG. 7. COMPARISON OF MOODY FORMULAE WITH I.P. MORRIS DATA [17]

TABLE 4

Installation	Runner diam. in.	Model diam. in.	Maximum model efficiency, per cent	Model efficiency at design phi, per cent	Efficiency increment, per cent	Expected maximum efficiency, per cent	Field test, per cent	Differences, test vs formula, per cent	Method of water measurement,
Queenston, No. 5...	112.5	36	91.0	91.0	2.3	93.3	93.3	0.0	Gibson
Queenston, No. 7...	113.1	36	90.9	90.8	2.4	93.2	93.8	+0.6	Gibson
Niagara No. 19.....	164.0	36	90.9	90.7	3.0	93.7	93.8	+0.1	Gibson
Spruce Falls........	93.0	16	85.7	85.4	5.1	90.5	90.5	0.0	Allen
Anson..............	92.0	16	86.7	86.0	4.7	90.7	90.5	—0.2	Allen
Holtwood No. 10 ..	149.5	16	87.0	87.0	5.5	92.5	92.8	+0.3	Allen
Cheat Haven.........	121.0	16	86.5	86.5	5.5	92.0	92.5	+0.5	Gibson
Rumfort Falls.......	85.0	32	90.2	90.1	2.1	92.2	92.8	+0.6	Allen
Norwood No. 3.....	170.5	16	87.5	87.5	5.5	93.0	92.5	—0.5	Gibson
Norwood No. 2.....	173.0	16	88.0	87.4	5.4	92.8	93.0	+0.2	Gibson
West Buxton..... ..	133.0	16	85.0	84.8	6.2	91.0	92.1	+1.1	Allen
Upper Tallassee.. ..	175.0	16	87.5	87.5	5.6	93.1	92.5	—0.6	Allen
Morony............	173.0	16	88.0	87.4	5.4	92.8	93.0	+0.2	Gibson
Chelsea...........	154.0	16	88.0	87.8	5.2	93.0	92.3	—0.7	Gibson
Light House Hill...	84.0	16	88.0	87.8	4.1	91.9	91.5	—0.4	Gibson
Paugan............	134.0	16	88.4	88.4	4.8	93.2	92.5	—0.7	Gibson
Dolby No. 8........	86.8	16	88.8	88.2	3.9	92.1	93.5	+1.4	Allen
Ghost Development.	106.0	16	88.0	87.4	4.6	92.0	92.5	+0.5	Gibson

Taken from page 729 of reference [17]

Fig. 3) but not enough to bring the formula into agreement with many other formulae. Angus [26] compared the efficiencies predicted by the formula with $n = 0.25$ and $n = 0.20$ for the tests of pumps for the Victoria Park Station in Toronto, Canada, see Table 5. There is very little choice between formulae for the four cases reported and the range of D/D' is too small to justify any real preference for one or the other.

Oesterlen's formula [11] was based on the studies by Hopf and Fromm [6] of friction loss in rough pipes. This led to $n = 0.314$ in the Moody formula. Oesterlen introduced the mechanical efficiency which evidently was assumed to be the same for both model and prototype because, in expanded form, the formula was published as

$$\eta = 1 - \left[1 - (\eta'/\eta_m)\right] (D'/D)^{0.314} \tag{38}$$

The formula was tested on the Lawaczeck turbine for the Lilla-Edet development with $\eta_m = 0.98$ apparently assumed. The values in Table 6 were taken from Oesterlin's paper.

It would appear that the rather low value of η_m offset the large value of n. The formula is a simpler version of a formula due to Haeger [10] but no wide acceptance of either formula has been mentioned in the literature.

The various formulae attributed to Pfleiderer have all been approximations or modifications of other formulae as noted in Table 2. The Gregorig formula [16] introduced the tail race velocity heads for both model and prototype. This was logical in that the model and prototype would be treated in the same way. The formula fit the tests of Riemerschmidt [14] on a Francis turbine ($D \cong 4$ inches) which was so small that the Reynolds number effects were important. Gregorig published data on three turbines, apparently two models and a prototype, but only one of the computed efficiencies could be verified from the data given in the paper.

In 1948, Mühlemann [35] published Ackeret's formula with the statement that it had been in existence since about 1930. The formula divided the losses equally between the kinetic and surface friction parts. Figs. 8 and 9 show the results of a large number of tests taken from the files of Ateliers des Charmilles, Geneva, Switzerland[1]. Points $1'$ and 1 were for a model and

[1] See also Figs. 2 and 3 of reference [38]

TABLE 5

PROTOTYPE AND MODEL PERFORMANCE -- VICTORIA PARK STATION [26]

PROTOTYPE					
PUMP NO.	CAPACITY	HEAD PER STAGE, H	SPEED	SPECIFIC SPEED	EFFICIENCY η
	gpm	feet	rpm		By test
1	8250	54	730	3350	0.902
2	16,500	82	750	3540	0.915
3	33,000	82	500	3350	0.930
4	20,600	135	750	2700	0.918

MODEL						
PUMP NO.	CAPACITY	HEAD PER STAGE, H'	SPEED	EFFICIENCY η'	D/D'	H/H'
	gpm	feet	rpm	By test		
1	4220	67.0	1200	0.900	1.48	0.81
2	4980	92.8	1500	0.896	1.88	0.88
3	4400	68.5	1200	0.896	2.63	1.20
4	4580	91.8	1200	0.907	1.94	1.47

COMPUTED PROTOTYPE EFFICIENCIES				
PUMP NO.	FORMULA 1 η_1	$\eta - \eta_1$	FORMULA 2 η_2	$\eta - \eta_2$
1	0.908	-0.006	0.905	-0.003
2	0.910	+0.005	0.907	+0.008
3	0.920	+0.010	0.914	+0.016
4	0.924	-0.006	0.921	-0.003
Average		+0.001		+0.005

Formula 1 . . . . . $(1-\eta)/(1-\eta') = (D'/D)^{0.25} (H'/H)^{0.10}$

Formula 2 . . . . . $(1-\eta)/(1-\eta') = (D'/D)^{0.20} (H'/H)^{0.10}$

TABLE 6

LAWACZECK TURBINE PERFORMANCE REPORTED BY OESTERLIN [11]

	MODEL 1	MODEL 2	PROTOTYPE
Runner diameters, mm	460	1000	6000
Head, meters	2.2	4	6.5
Maximum efficiency by test	0.839	0.862	0.917
η computed from Model 1	-----	-----	0.918
η computed from Model 2	-----	-----	0.913

prototype respectively of a 47,000 horsepower Francis unit for which exceptionally good test data were available ($D'/D = 1/4$; $H'/H = 1/20$; $\eta'/\eta = 0.894/0.918$; $\eta'_m = 0.994$). The Ackeret formula predicted an efficiency $\eta = 0.921$ for the prototype which was only 0.003 higher than the measured value. Figs. 7, 8 and 9 illustrate clearly the dependence of any single conversion formula on individual fabrication and testing procedures for laboratory models. The Ackeret formula would not fit the I.P. Morris data any better than the Moody formula would fit the Charmilles data. Also, Figs. 7 and 8 show a scatter of about ±0.01 whereas Fig. 9 shows a scatter of at least ±0.02 for which the dashed lines were drawn.

Medici [31, 38] appears to have been the first to publish a conversion formula specifically for pumps. As mentioned in Section E.3., the difference in numerical values between a corresponding pump and turbine formula is very small for most cases of practical interest, see also Fig. 2. Medici [38] reported laboratory tests of turbines over a range of scale ratios not exceeding 1/2.5. He stated, "Several scales were adopted, and care was taken that the models should exactly reproduce the original even to the roughness of the wetted surfaces." He revised Mühlemann's results, mentioned above, and showed that his own and the Charmilles tests defined a single curve. Fig. 4, taken from Medici's paper [38], shows values of the f - function for use in his formula. Unlike the efficiency-conversion formulae, Medici's formula estimates the discharge or capacity of a prototype turbine from data obtained from a model test. He stated, "The relationship applies also to water pumps, the only variant being

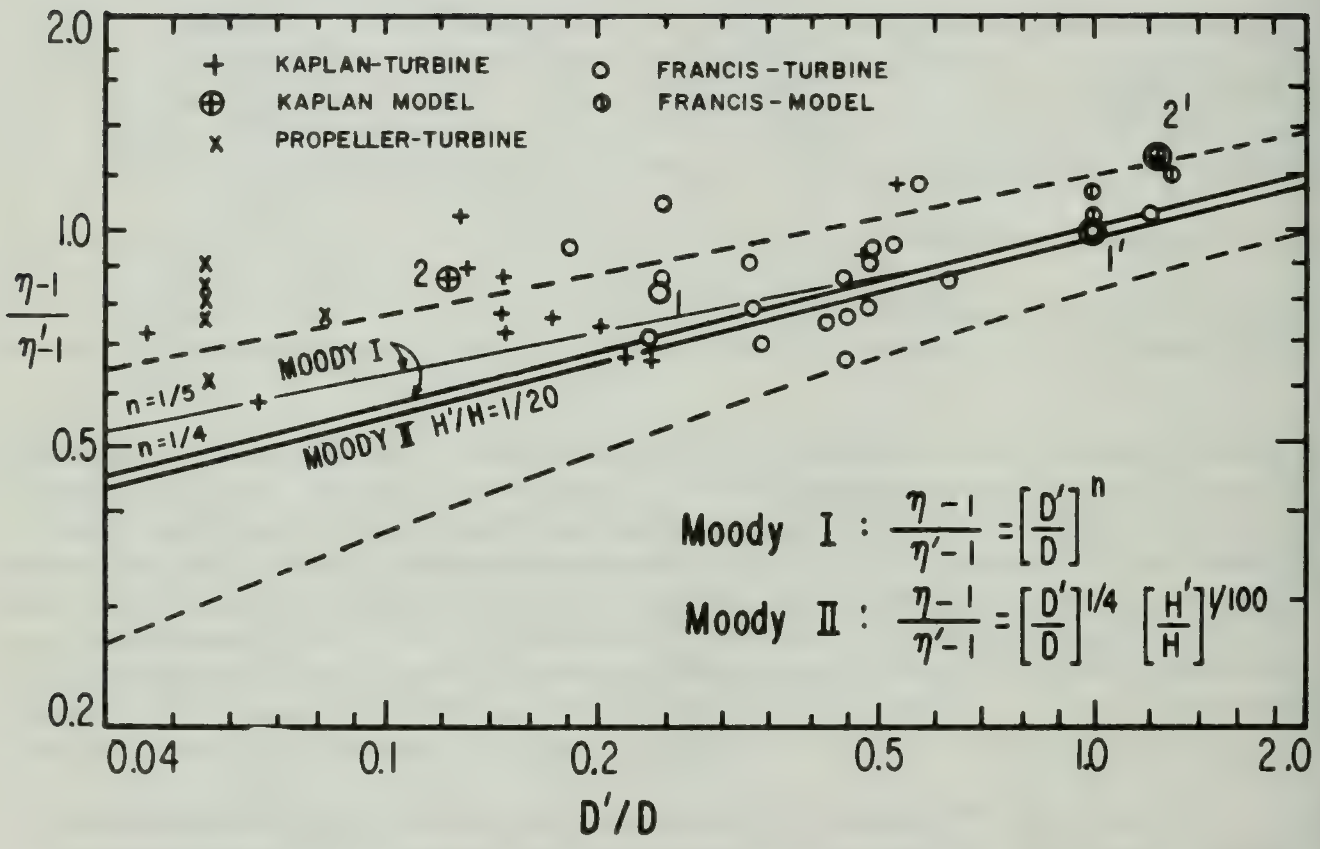


FIG. 8. COMPARISON OF MOODY FORMULAE WITH CHARMILLES DATA [35]

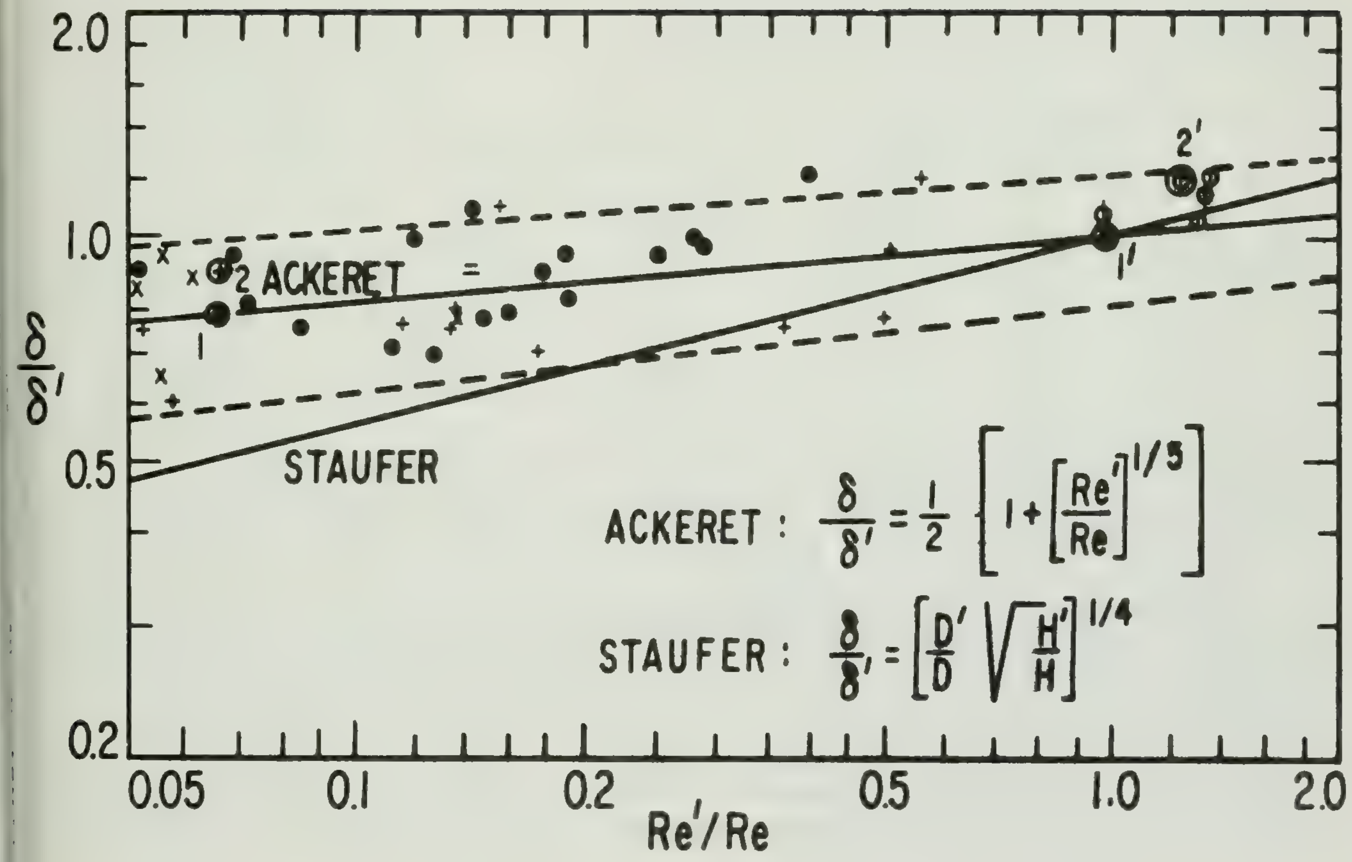


FIG. 9. COMPARISON OF STAUFER AND ACKERET FORMULAE WITH CHARMILLES DATA [35]

the numerical value of the f - coefficient". Medici described the Charmilles turbine tests as covering the range of specific speeds[1] from 26 to 240 and the range of power from 250 KW to 40,000 KW.

In 1951, Rütschi [41, 45] reported extensive tests on an entire line of single-stage pumps. Figure 10 shows the three separate curves Rütschi obtained by plotting the hydraulic efficiencies against impeller diameter, D, and Fig. 11 shows how the same data reduced to a single curve when plotted against the impeller eye diameter, D_o. In an earlier paper [18], Rütschi reported tests on small multi-stage pumps as shown in Table 7.

TABLE 7

TESTS ON SMALL MULTI-STAGE PUMPS REPORTED BY RÜTSCHI [18]

PUMP	CAPACITY	TOTAL HEAD	SPECIFIC SPEED	SPEED	η	η_h
	gpm	feet		rpm		See Note 2
3 stage	150	294	4180	2900	0.755	0.88
9 stage	286	430	4960	1460	0.81	0.907
4 stage	1080	378	6100	1470	0.825	0.905

He estimated that the optimum attainable hydraulic efficiency for pumps with fixed guide vanes and specific speeds of 6200 or less should be about 0.905 and for volute, mixed flow, and axial flow pumps the optimum hydraulic efficiency[3] should be about 0.945. The latter value was used as a normalizing factor for the data in Fig. 11 to prepare Fig. 5 under the assumption that D_o = 2500 millimeters (8.2 feet) represented a limit above which no further increase in efficiency with increase in size should be expected. The hydraulic efficiencies

[1] $n_s = N \sqrt{\text{Horsepower}}/H^{5/4}$

[2] Estimated from measured and computed losses.

[3] Pantell [50] criticized the hydraulic efficiencies used by Rütschi as being too low. It seems likely that very large pumps of the best design and construction could have hydraulic efficiencies higher than 0.945.

of pumps used for cold water should be in direct proportion to the ordinates (values of f) of the curve in Fig. 5. The extrapolated portion was tested against published efficiencies for the Etzelwerk [39] five-stage 20,220 horsepower pumps as follows:

$$D_o' = 300 \text{ millimeters} = 11.8 \text{ inches}; \quad \eta' = 0.855$$

$$D_o = 900 \text{ millimeters} = 35.4 \text{ inches}; \quad \eta = 0.870$$

Rütschi assumed $\eta_m' = 0.990$ and $\eta_m = 0.998$ which led to $\eta_h' = 0.864$ and $\eta_h = 0.872$. The ratio of the f-values from Fig. 5 agreed exactly with the ratio of the hydraulic efficiencies. Thomann [39] reported increments in efficiency for the Etzelwerk pumps as

Moody,	$\Delta\eta$ =	0.035
Ackeret,	$\Delta\eta$ =	0.036
Sulzer,	$\Delta\eta$ =	0.025
Actual test,	$\Delta\eta$ =	0.015

but no details of the computation were given.

Garve [53] proposed a family of curves to replace the single f-function (Fig. 5) of Rütschi and a method for applying the Rütschi formula to turbines. The effect of the family of curves was to shift the level of the hydraulic efficiency up or down depending on the design and construction of the pumps in question. In a discussion of these proposals, Rütschi [53] pointed out that his formula had yet to be proved applicable to hydraulic turbines and that the family of curves would introduce a complexity of doubtful value since it was unlikely that efficiency conversion would be of much interest unless the pumps were of excellent design and construction. He stated that the pumps he had used in the development of the formula [41, 45] were of very good hydraulic design. The wheel castings were clean but not better in finish than would be customary in any scientific investigation.

Pantell [50] stated that he had developed a conversion formula specifically for pumps, similar to Medici's [31] 1943 formula, as early as 1942 in a series of lectures sponsored by the German Society of Engineers (VDI). He examined Rütschi's results using the Blasius [4] formula for friction loss in smooth pipes and a formula due to K. Fromm [6] for rough pipes. The Blasius formula showed the friction to be inversely proportional to the one-fourth power

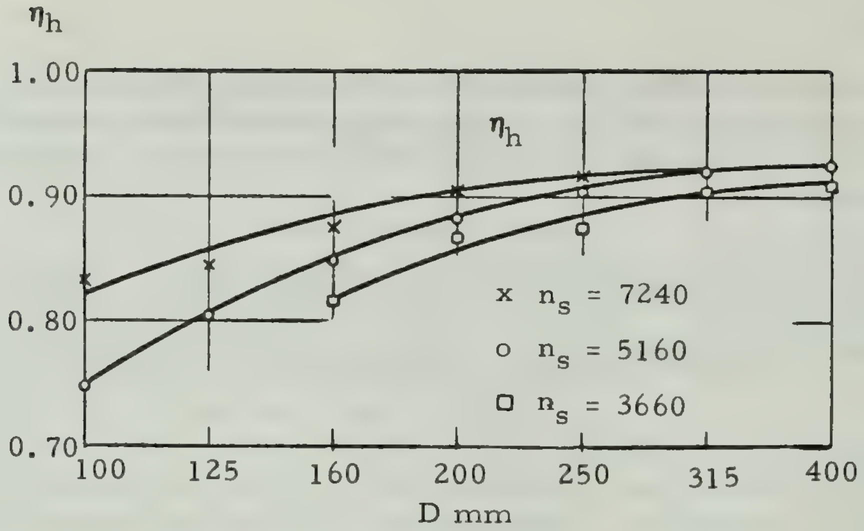


FIG. 10. HYDRAULIC EFFICIENCY AS A FUNCTION OF IMPELLER DIAMETER. RÜTSCHI [41]

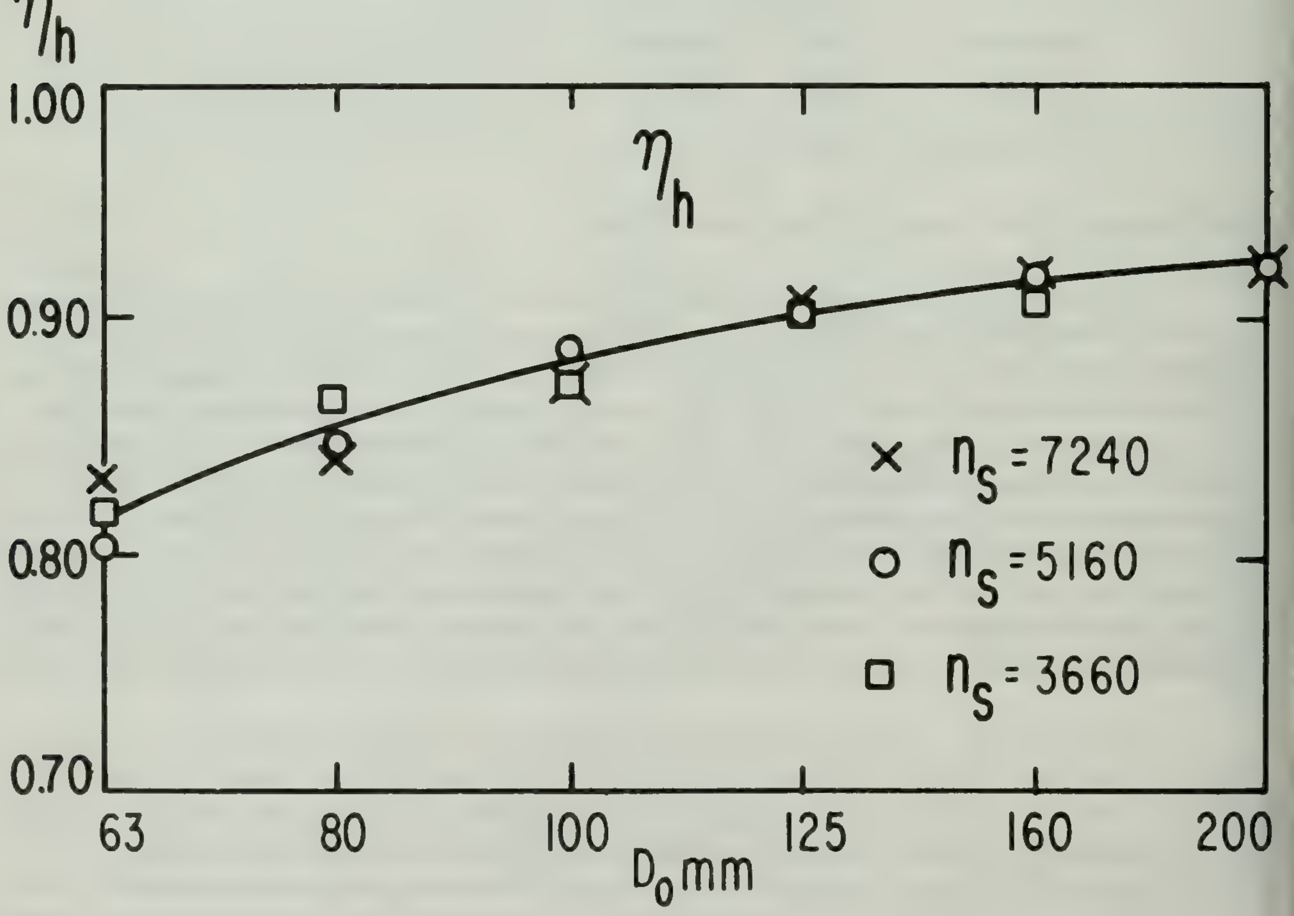


FIG. 11. HYDRAULIC EFFICIENCY AS A FUNCTION OF IMPELLER EYE DIAMETER. RÜTSCHI [41]

of the Reynolds number which under the assumption that both the model and the prototype pumps would be tested at the same speed and with the same fluid, was equivalent to the ratio of the impeller diameters squared. The conversion formula became

$$\left[(1-\eta_h)/(1-\eta_h')\right](\eta_h'/\eta_h) = \sqrt[4]{Re'/Re} = \sqrt[4]{(N'D'/ND)(D'/D)} = \sqrt{D'/D} \qquad (39)$$

which was used to compute curve a of Fig. 6 with $\eta_h = 0.925$ for an impeller diameter, D = 400 millimeters (16 inches), from Rütschi's tests. Curve b of Fig. 6 was the middle ($n_s = 5160$) curve of Fig. 10 and showed that Eq. (39) did not approximate the losses in impellers of small diameter. To correct for the increased importance of the surface roughness in impellers of small diameter, Fromm's formula [6] was used to obtain

$$\left[(1-\eta_h)/(1-\eta_h')\right](\eta_h'/\eta_h) = \left[(k_F/k_F')(D'/D)\right]^{0.314} \qquad (40)$$

Hydraulic efficiencies taken from curve b of Fig. 6 were used in Eq. (40) to compute the values of k_F, also shown in Fig. 6, with $k_F = 1$ and $\eta_h = 0.92$ at D = 350 millimeters where curve b joined curve a for smooth surfaces. Figure 6 may apply to the model, prototype, or both depending only on the impeller diameter, D.

Pantell stated that considerable experience would be required to obtain good results with the formula for rough surfaces. He questioned the computed volumetric efficiencies for Rütschi's small pumps on two counts; geometrical similarity in the wearing ring clearances probably could not have been maintained and accurate measurements of the clearances would have been very difficult to make. Both, according to Pantell, might have led to too low values of the hydraulic efficiency. He felt that the simplifying assumptions made in the development of most of the formulae would limit their applicability to the vicinity of the maximum efficiency and even this could scarcely be predicted within ±2% for the prototype pump or turbine.

Krisam [51] pointed out that Rütschi's formula included a measure of the actual sizes of pumps under consideration whereas most other formulae contained only the scale ratio. The conversion from D' = 100 millimeters to D = 200 millimeters would be quite different from the conversion from D' = 400 millimeters to D = 800 millimeters although the scale ratio was two to one in

each case. Figure 12 shows efficiencies reported by Krisam for a geometrically similar series of seven multi-stage high-pressure pumps, specific speed = 3510, together with efficiencies predicted by the Rütschi formula computed from the smallest pump as a model of all the others. Krisam attributed the lack of agreement between the computed hydraulic efficiencies and the actual test points to the friction losses in the narrow impellers and complicated vaned crossover passages of the multi-stage pumps which probably were higher than in the single-stage volute-cased pumps on which the formula was based. There were small differences in the efficiencies predicted by the several Rütschi formulae for the f-function including that of the German Society of Engineers [47] (see Table 2). Curve a in Fig. 12, which is about one-half of one percent in efficiency higher than that designated by the open circles, represents the upper limit of the several formulae. Curve b of Fig. 12 shows the Rütschi formula applied to Krisam's data with the largest pump as model and the smaller pumps as prototypes. This indicates the increasing importance of friction in the smaller multi-stage pumps.

The German Society of Engineers (VDI) [47] adopted Rütschi's method with relatively minor modifications. Small corrections for speed and impeller diameter were introduced and the equation to represent Rütschi's f-function was changed, presumably to improve the degree of approximation to the curve in Fig. 5. The details of these changes are given in Table 2.

Rotzoll [58] made very extensive tests of a single-stage pump for which D_o = 100 millimeters, (3.94 inches) and D = 199 millimeters (7.84 inches). By using oil and water at different temperatures and by changing the pump speed, the range of Reynolds numbers shown in Fig. 13 was covered. The legend of Fig. 13 describes the various curves and how they may be used. Rütschi [59] analyzed the characteristics of seventeen pumps and turbines covering a very wide variety of sizes and types. He found that the hydraulic efficiencies correlated well when plotted against a Reynolds number based on the eye diameter of a pump or the throat diameter of a turbine as shown by the dot dash curve in Fig. 14. Rotzoll's curve of maximum hydraulic efficiencies was used as a guide to establish a family of curves, dashed lines in Fig. 14, which would correct for changes in Reynolds number due to changes in speed or fluid viscosity. A discussion of the Rütschi-Rotzoll analysis and conclusions was given by Pfleiderer [63], pp. 170-174. In connection with Reynolds number effects, Tetlow [30], p. 123, stated "Experimental tests indicate, in no uncertain manner, that the larger the pump, the smaller is the proportional effect of viscosity."

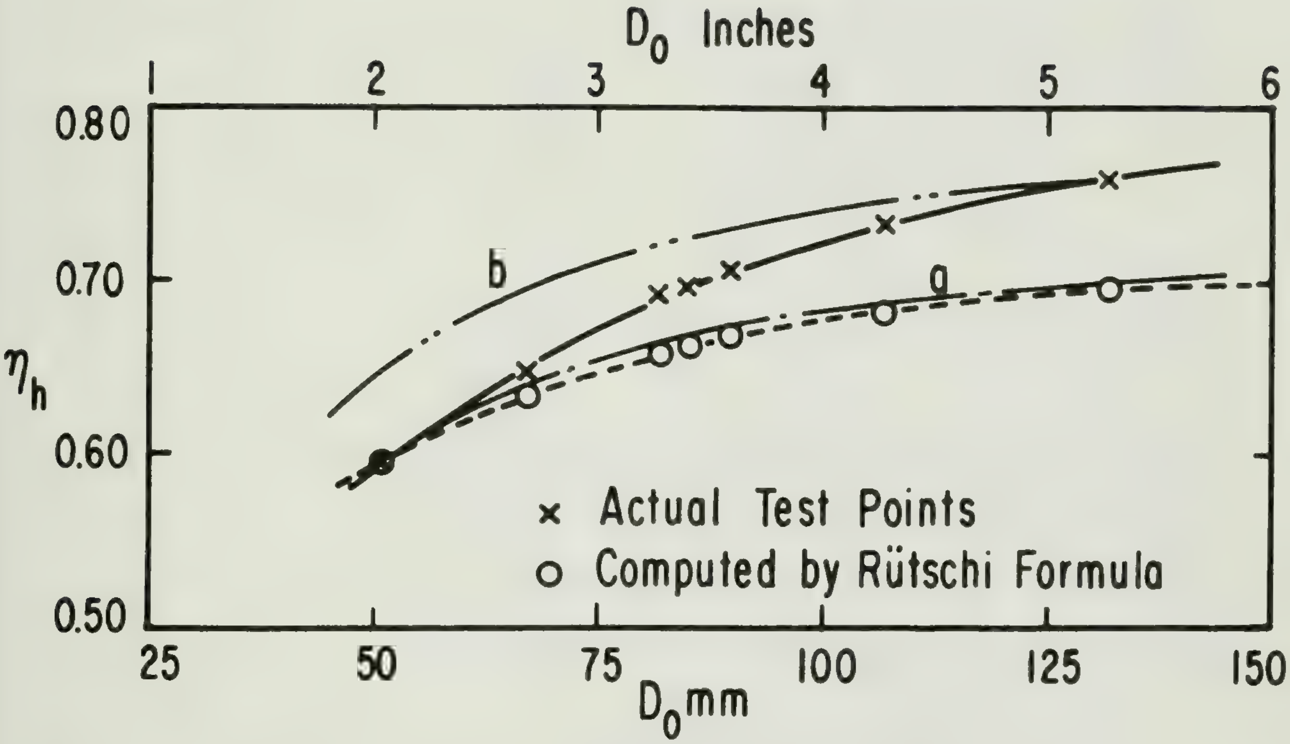
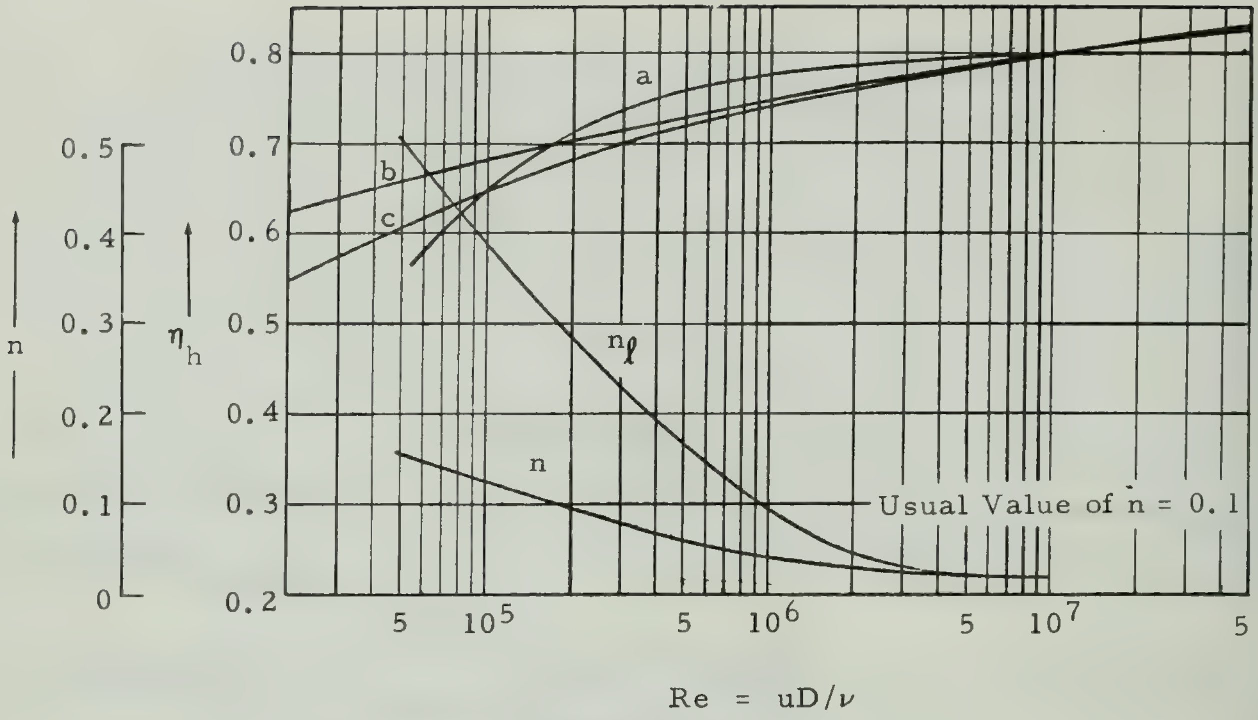


FIG. 12. RÜTSCHI FORMULAE APPLIED TO MULTI-STAGE PUMPS (n_s = 3510) BY KRISAM [51]



Curve a, measured values of maximum hydraulic efficiency, η_h

Curve b, η_h computed by Pfleiderer's formula

$$(1-\eta_h)/(1-\eta_h') = (Re'/Re)^n$$

Curve c, η_h computed by Ackeret's formula

$$(1-\eta_h)/(1-\eta_h') = 0.5 + 0.5\,(Re'/Re)^{2n}$$

Curve n, values of n' and n based on Re = 10^7 for use in

$$(1-\eta_h)/(1-\eta_h') = (Re'/10^7)^{n'}/(Re/10^7)^n$$

Curve n_ℓ, local values of n for use in

$$(1-\eta_h)/(1-\eta_h') = (Re')^{n'_\ell}/(Re)^{n_\ell}$$

Read n_ℓ at Re and n_ℓ' at Re'

FIG. 13. HYDRAULIC EFFICIENCIES AND VALUES OF n ROTZOLL [58]

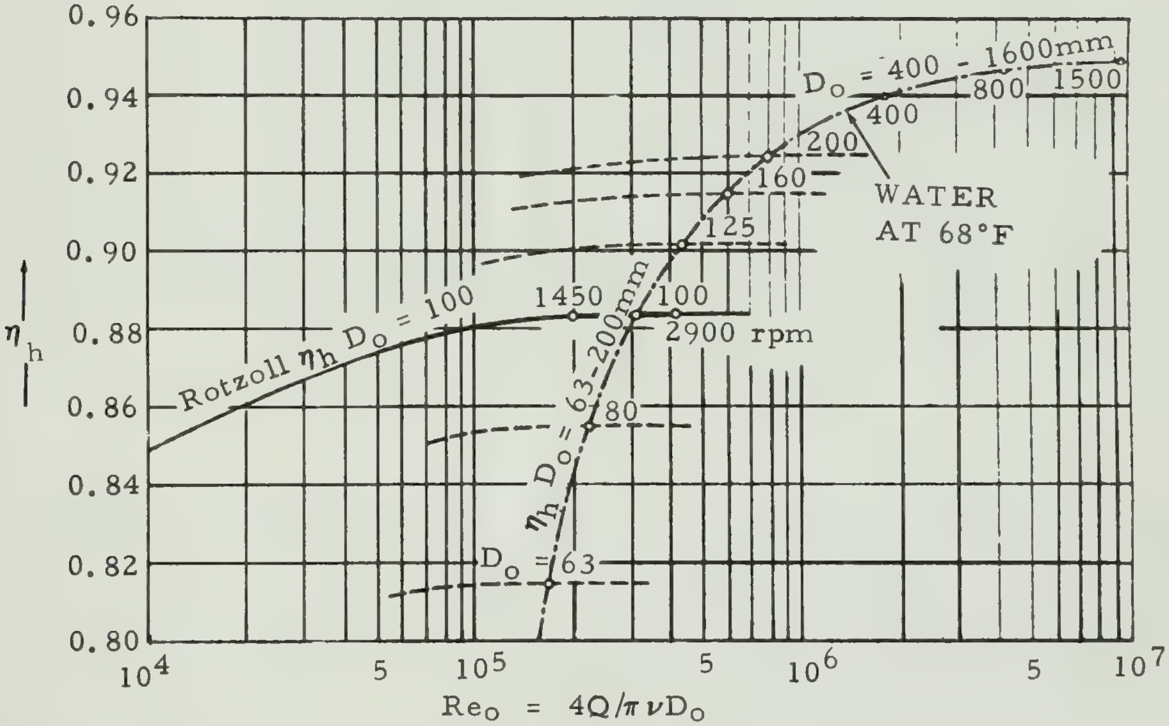


FIG. 14. HYDRAULIC EFFICIENCIES OF PUMPS. RÜTSCHI [59]

The two turbine formulae proposed by McDonald [40] were based on friction formulae due to Prandtl and Schlichting [55]. The smooth-surface formula was based on a flat plate formula, [55], p. 540, and the rough-surface formula was based on a pipe friction formula, [55], p. 525. The dimensionless coefficients c and f must be determined by experiment.

2. Formulae for Predicting Over-all Performance

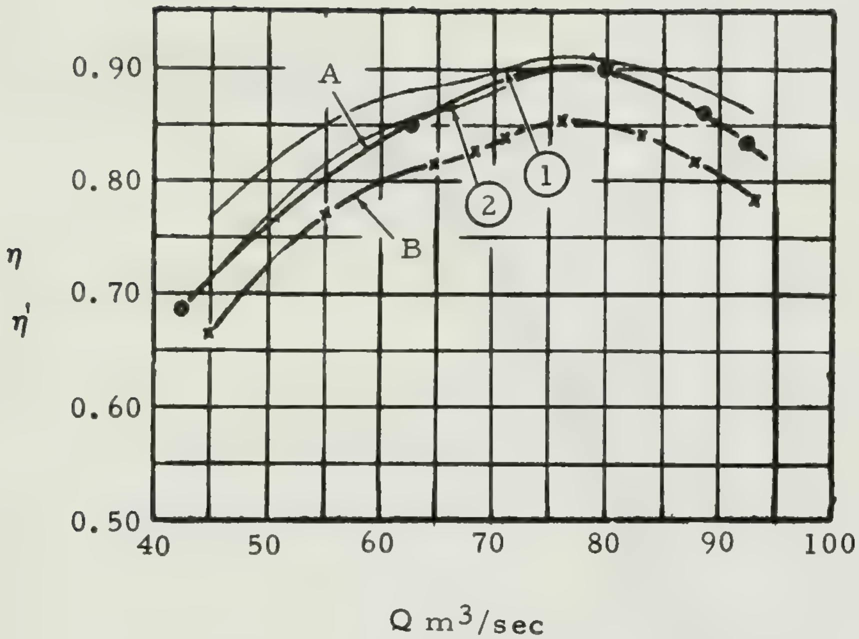
Although most of the formulae have been developed for the maximum efficiency point only, various attempts have been made to extend some of them to the full operating range. The Moody formulae have been used in this way [23, 24] but with some loss of accuracy away from the best efficiency point. An early attempt to cover the over-all performance was made by Miyagi [19] who inserted a correction term based on discharge in a formula of the Reynolds number type. Fuchizawa [42] assumed that the friction losses would be constant over the entire operating range, that the friction losses could be treated as for an equivalent rough pipe, and that the eddy (kinetic) losses would be negligible at maximum efficiency. Rearrangement of the formula yields

$$(\delta-\delta')/\delta_o' = 1 - \left[(\epsilon/\epsilon')(D'/D)\right]^{0.25} \tag{41}$$

wherein the subscript o refers to the maximum efficiency point. The formula with $\epsilon = \epsilon'$ was in excellent agreement with results of tests of the Raanaasfoss, Norway, No. 1 turbine, as shown in Fig. 15, but when used to predict the efficiencies of two other Francis turbines, there were discrepancies of as much as three per cent over most of the operating range. Figure 16 shows four different formulae, including that of Fuchizawa, applied to tests of three identical Francis turbines. No formula could fit the tests of the three units as they differed by as much as two per cent but the Fuchizawa formula was the best of the four.

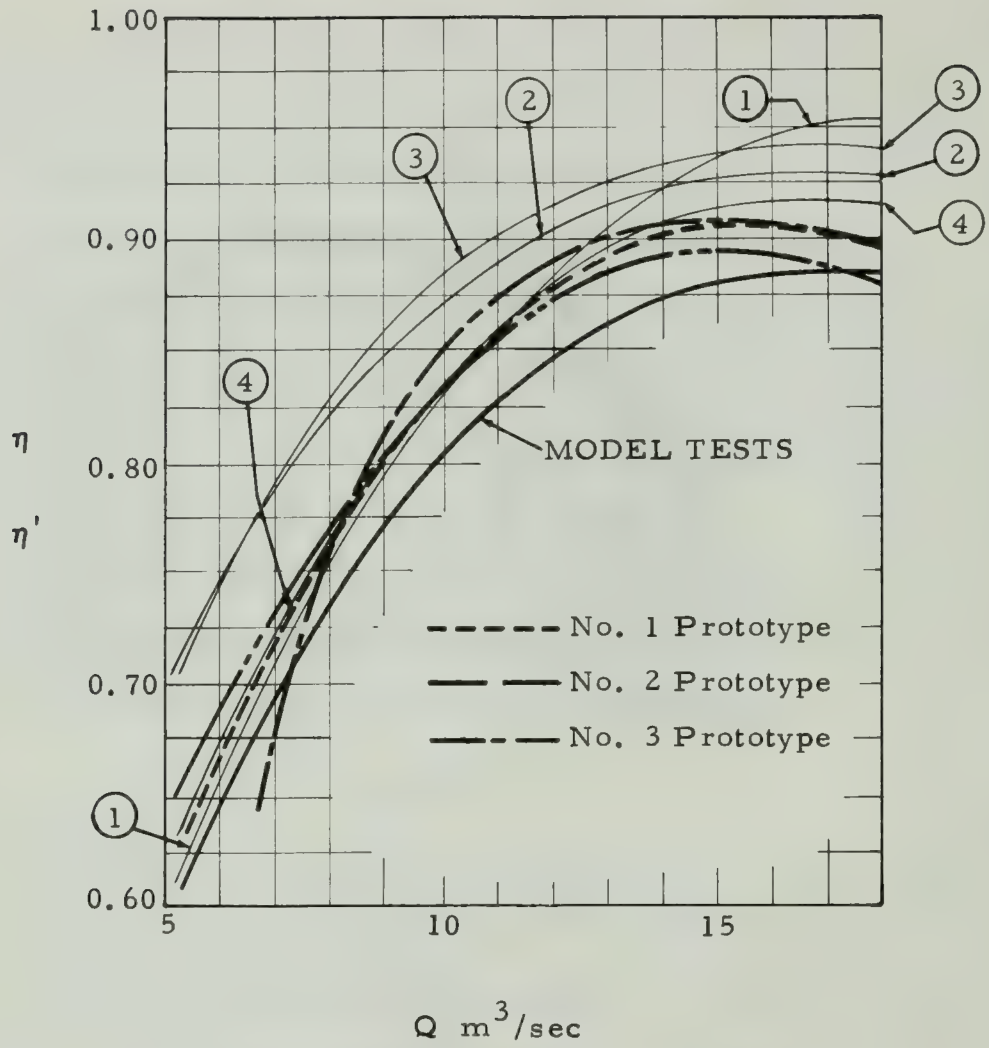
Yamazaki [46] derived two formulae for turbines under the assumption that the losses would be partly friction and partly eddy (kinetic) losses. The surface friction was assumed to be inversely proportional to a Reynolds number raised to the one-fifth power. The kinetic losses were approximated by analogy to form-drag coefficients for flat plates at an angle of attack, α, to the approaching flow,

$$C_d = C_{d\,max}\,(1 - \cos\alpha) \tag{42}$$



Curve A.	Test of Prototype Turbine
Curve B.	Test of Model Turbine
Curve 1	Komoriya - Hatori Formula
Curve 2	Fuchizawa Formula

FIG. 15. RAANAASFOSS POWER STATION NO. 1. UNIT. [42]



Curve 1. Miyagi Formula

Curve 2. Komoriya - Hatori Formula

Curve 3. Yamazaki's Second Formula

Curve 4. Fuchizawa Formula

FIG. 16. MEASURED AND PREDICTED EFFICIENCIES [42]

wherein,

C_d = drag coefficient at angle of attack α

$C_{d\,max}$ = drag coefficient at $\alpha = 90°$

Three additional assumptions were:

(1) At $Q \cong 0$, the friction loss could be neglected.

(2) At $Q = Q_o$, the kinetic loss could be neglected.

(3) $\cos \alpha \cong (Q/Q_o)$ an assumption also attributed to Miyagi.

A summation of the losses led to the formula in Table 2 designated as "First Formula." Yamazaki did not indicate how to use the formula when $Q > Q_o$. By assuming that $(1 - \cos \alpha)$ could be approximated by a parabola $[(Q_o - Q)/Q]^2$, Yamazaki converted the first formula to the form designated as "Second Formula" in Table 2. Figures 17 and 18 taken from reference [46] show efficiencies predicted by both formulae compared with efficiencies by actual test for two Francis turbines. Both figures show a scatter of more than three per cent in efficiency for the test points of the prototype turbines as well as an apparent shift of the maximum efficiency point for each prototype turbine to smaller discharges than predicted by the models. The second formula also appears as Curve 3 in Fig. 16.

The analyses of Hirotsu [48, 49] were carried out first for Francis turbines and applied to units of the Senju and Nezama Power Stations as shown in Fig. 19. He computed the disk friction and found it to be negligibly small in each case. The computed leakage flows indicated a volumetric efficiency for Senju of 0.99 and for Nezama, 0.986. On a basis of these computations, Hirotsu stated he felt justified in assuming the hydraulic efficiency to be essentially the same as the over-all efficiency for both turbines. He estimated the friction losses as being inversely proportional to Reynolds number raised to the one-fourth or one-fifth power and found that the one-fifth exponent lowered the predicted efficiency about one per cent as compared with that predicted by the one-fourth exponent. At part load operation, Hirotsu defined a "shock" velocity, C_t, given by

$$C_t / \sqrt{2gH} = a_1 \eta_h - a_2 + a_3 q \qquad (43)$$

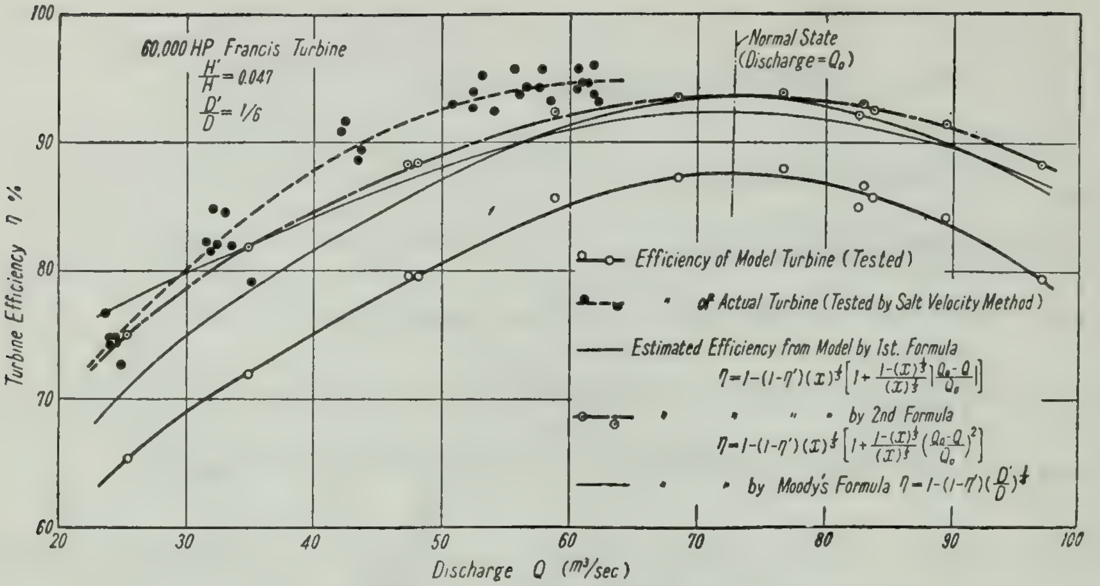


FIG. 17. TURBINE EFFICIENCY CONVERSION
BY YAMAZAKI FORMULAE [46]

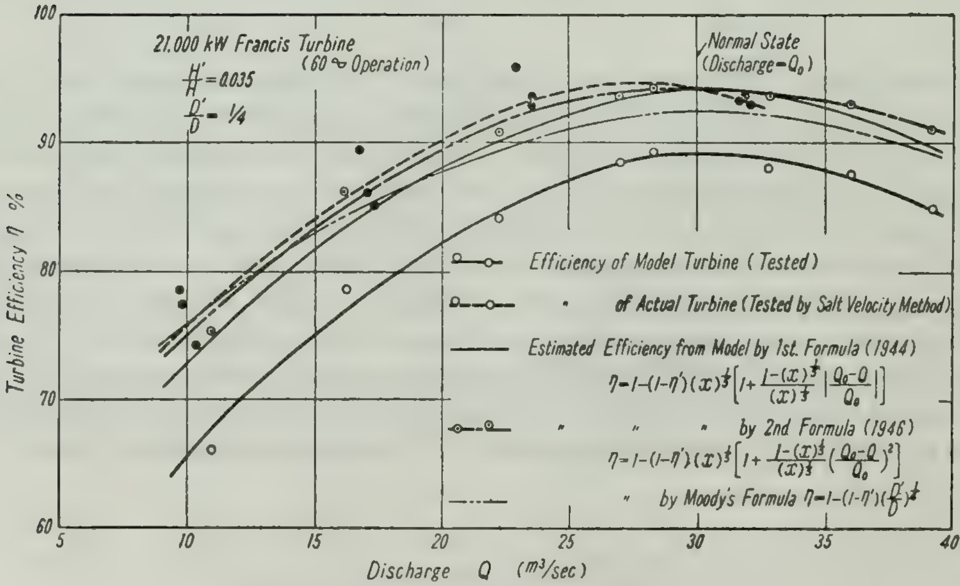
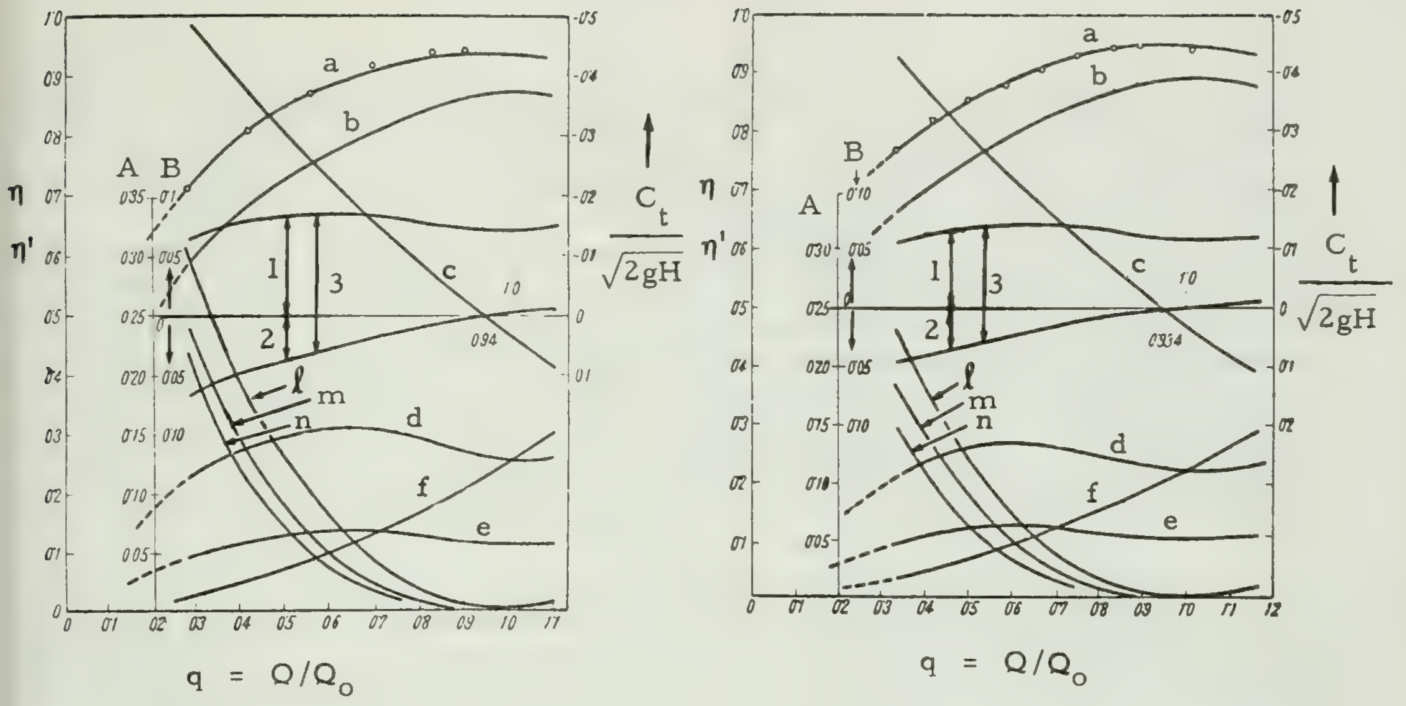


FIG. 18. TURBINE EFFICIENCY CONVERSION
BY YAMAZAKI FORMULAE [46]



(a) Senju Power Station

(b) Nezama Power Station

Curve a.	Average of Tests of Prototype Turbine	
Curve b.	Tests of Model Turbine	
Curve c.	$C_t/\sqrt{2gH}$; C_t = Shock Velocity	
Curve d.	$(1-\eta') - (C_t'^2/2gH')$	Use Scale A
Curve e.	$(1-\eta) - (C_t^2/2gH)$	Use Scale A
Curve f.	$(1-\eta'_{max})\, q^{7/4}$, Miyagi's friction loss	Use Scale A
Curve ℓ	$C_t'^2/2gH'$ — Hirotsu's shock loss	Use Scale A
Curve m	$C_t^2/2gH$ — Hirotsu's shock loss	Use Scale A
Curve n	$(1-\eta')(1-q)$ Yamazaki's shock loss	Use Scale A

1 Friction Loss in Model minus Friction Loss in Prototype

2 Shock Loss in Model minus Shock Loss in Prototype

3 Total Efficiency Conversion from Model to Prototype

(Items 1–3: Use Scale B)

FIG. 19. LOSSES IN FRANCIS TURBINES COMPUTED BY HIROTSU [48]

wherein, $q = Q/Q_o$ and a_1, a_2, and a_3 were constants. Hirotsu made use of the maximum efficiency point, where he assumed the shock loss to be negligible, and the condition of $q \cong 0$, where the efficiency was very small and $C_t^2/2gH \cong 1$, to solve for the constants in Eq. (43). The values obtained were $a_1 = 1$, $a_2 = 0.5$, and $a_3 = 1 - (\eta_o/2)$ under the further assumption that the hydraulic and over-all efficiencies could be considered equal. The same analysis served for both model and prototype. The equation listed in Table 2 resulted from a summation of the friction and shock (kinetic) losses. In extending the analysis to propeller and Kaplan turbines, Hirotsu considered the circulation in the wake from the runner blades as well as a shock loss computation based on the velocity diagrams but concluded that the formula for Francis turbines could be modified as shown in Table 2 to cover the three classes of machines.

Hutton [52] analyzed the runner and draft-tube losses of a Kaplan turbine operating on the cam curve, i. e., both runner and guide vanes positioned for maximum efficiency at any load condition. Losses in the scroll case, guide vanes, and other passages leading to the runner were assumed negligibly small. The results of the analysis are shown in Figs. 20 and 21 taken from reference [52]. Figure 20 shows that, at the point of best efficiency ($Q/Q_o = 1$), thirty per cent of the losses were kinetic and seventy per cent friction. At this point the formula should read

$$\delta/\delta' = 0.3 + 0.7\,(R'e/Re)^{0.20} \tag{44}$$

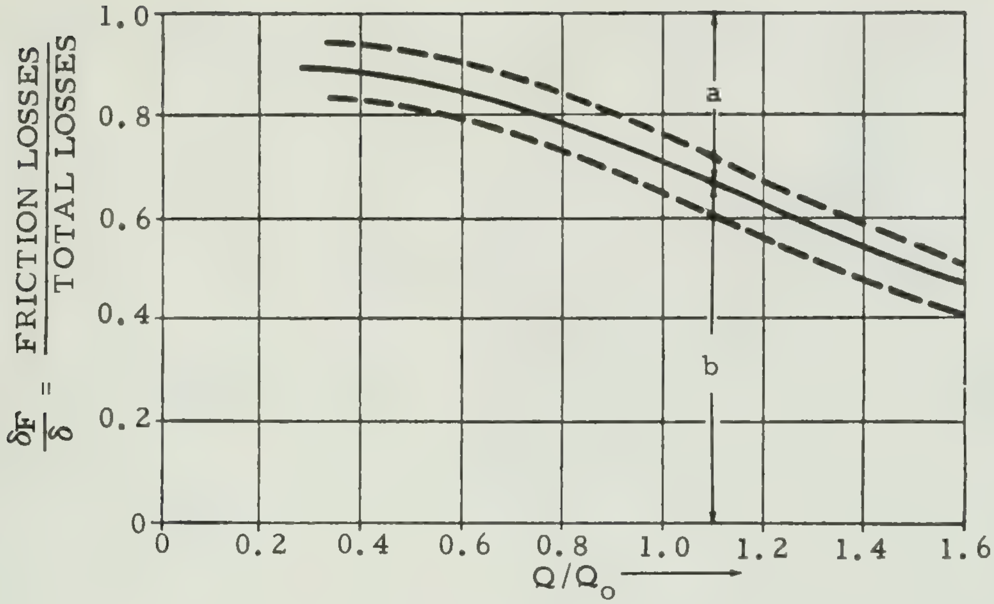
For any other operating point, the numbers 0.3 and 0.7 should be replaced by values of a and b respectively, read from Fig. 20. Figure 21, taken from reference [52], shows the formula applied to the operating ranges of two turbines. The predicted efficiencies at part load appear to be very good. When applied to the maximum efficiency only, Hutton's formula gave results similar to the Moody 1942 formula as shown in Fig. 22, where the comparison has been made for the case of $(H/H') = 10$. See also Fig. 3.

According to the affinity laws, for geometrically similar pumps and turbines

$$Q/ND^3 = C_1 = \text{a constant} \tag{45}$$

and

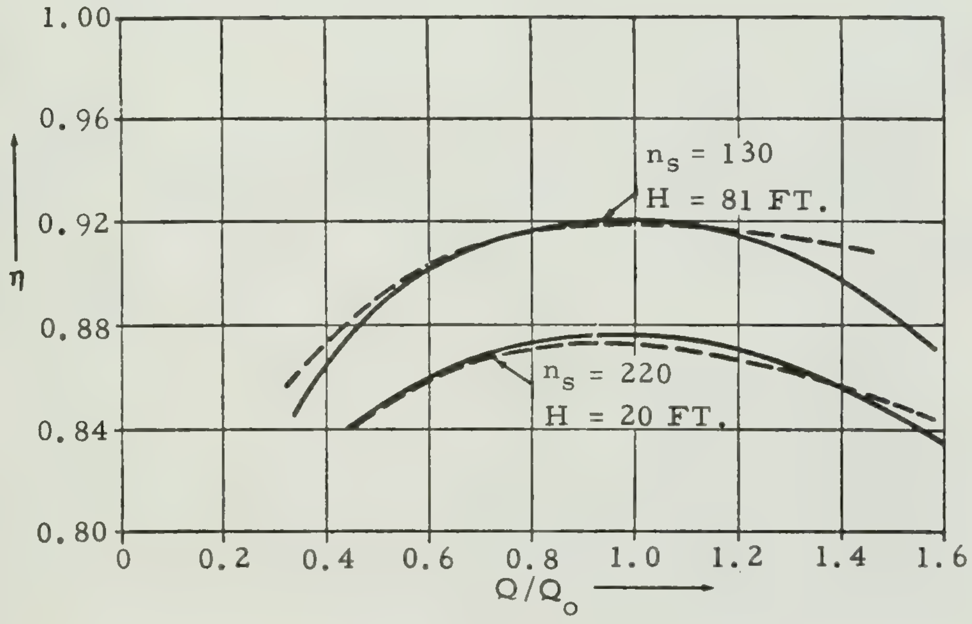
$$H/(ND)^2 = C_2 = \text{a constant} \tag{46}$$



——————— Mean curve.

– – – – – – – Possible limits of theory.

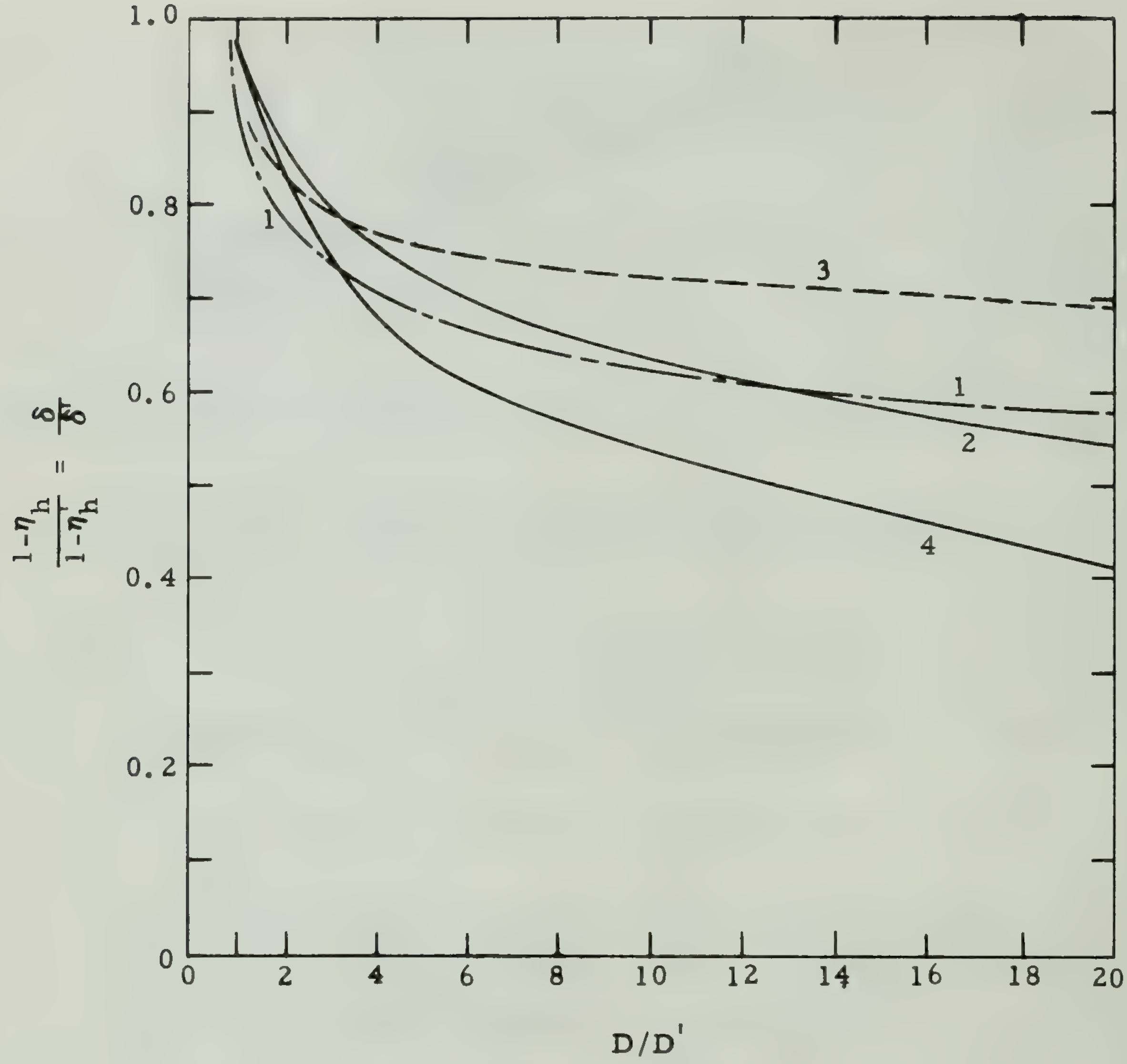
FIG. 20. VARIATION OF FRICTION/TOTAL LOSS RATIO WITH FLOW, HUTTON [52]



——————— Measured

– – – – – – – Computed

FIG. 21. MEASURED AND COMPUTED EFFICIENCIES, HUTTON [52]



1. Hutton $\delta/\delta' = 0.3 + 0.7\,(Re'/Re)^{0.20}$

2. Moody $\delta/\delta' \cong (D'/D)^{0.20}$

3. Ackeret $\delta/\delta' = 0.5 + 0.5\,(Re'/Re)^{0.20}$

4. Moody $\delta/\delta' \cong (D'/D)^{0.25}\,(H'/H)^{0.01}$

FIG. 22. COMPARISON OF TURBINE FORMULAE (H/H') = 10

Since the power must be proportional to the product QH, there could be no change in efficiency between a prototype and a geometrically similar model operated under dynamically similar conditions if Eqs. (45) and (46) were satisfied exactly. In practice, the affinity laws do not hold strictly so Itaya, Tejima, and Nishikawa [61] suggested that, in addition to the usual scale ratio, a velocity ratio and a head ratio might be introduced to account for lack of complete dynamic similarity in the actual flow phenomena. The authors properly pointed out that friction tests on circular pipes or flat plates should not be expected to provide accurate composite friction coefficients for the complicated flow passages of a pump or turbine. Separate friction coefficients for the model and prototype were introduced to remedy this defect. The various coefficients were to be determined by a series of tests on two or more models of different sizes or by a series of tests on a single model at different speeds or by a combination of both. Such a series of tests has rarely, if ever, been available although it would provide a wealth of information on which to base a prediction of prototype efficiency. The formula credited to Itaya, Tejima, and Nishikawa in Table 2 was selected as a compromise among the several proposed. The authors recommended their methods as applicable to the useful operating range of centrifugal pumps and turbines but restricted them to machines having smooth surfaces only.

Kovats [66] has suggested that the principal losses in a pump might be evaluated at a sufficient number of points throughout the useful operating range to obtain the equivalent of a computed efficiency curve. He listed formulae for computing friction, diffusion, and leakage losses, each of which were dependent on experimentally determined coefficients. The geometry of the flow passages and the dynamics of the actual flow phenomena in both pumps and turbines are so complicated that no accurate evaluation of all of the individual losses appears to be practicable in the near future. Flügel [54] also discussed methods of evaluating some of the losses in pumps and turbines but presented few numerical data on which computations might be based.

3. Discussion of Methods for Estimating Surface Friction

The major loss in a pump or turbine operated at or near the point of maximum efficiency appears to be surface friction. Nearly all investigators have used some function of Reynolds number to evaluate it. The Blasius formula for smooth pipes [4] showed the friction loss to be proportional to $1/Re^{0.25}$ provided the Reynolds number, $Re = Vd/\nu$, did not exceed 100,000. An empirical formula

$$f = 0.0000535 + 0.7257/(\log Re)^{2.297} \qquad (47)$$

was found to fit the smooth pipe data of Freeman [28] in the range Re >80,000. Friction losses in rough pipes at Reynolds numbers large enough to insure complete turbulence have been given by the Prandtl -von Karman-Nikuradse equation[1]

$$1/\sqrt{f} = 2 \log \left[3.7/(\epsilon/d)\right] \tag{48}$$

which has been approximated by[1]

$$f \cong 0.0967 \, (\epsilon/d)^{0.224} \tag{49}$$

The Colebrook formula [21], which applies to all types of turbulent flow in pipes, is not in a form convenient for application to pumps and turbines.

Formulae for friction loss on flat plates have been used by several authors. These are known as boundary layer equations. For a laminar boundary layer, the friction has been found to be proportional to $1/Re^{0.5}$ where $Re = V\ell/\nu$. The velocity, V, was that of the free stream outside the boundary layer and ℓ the length of the plate. For a turbulent boundary layer on a smooth plate, the exponent 0.5 has been found to reduce to 0.2 over a limited range of Reynolds number. An empirical equation applicable over a rather wide range of Reynolds number has been given by Schlichting [55] as

$$f = 0.455/(\log Re)^{2.58} \tag{50}$$

Usually the upstream portion of the boundary layer on a flat plate parallel to the flow has been found to be laminar and the downstream portion turbulent. The formulae recommended for rough plates generally have contained logarithmic terms raised to some power n. An approximate value n = -2.5 has been used but no single value applies to all cases. Details of pipe friction and boundary layer computations may be found in reference [55]. The foregoing have been included to show that either pipe friction or flat plate analogies could have been used to construct almost any of the formulae in Table 2. It should be remembered that a ratio of Reynolds numbers may simplify considerably under the circumstances present in hydraulic machinery.

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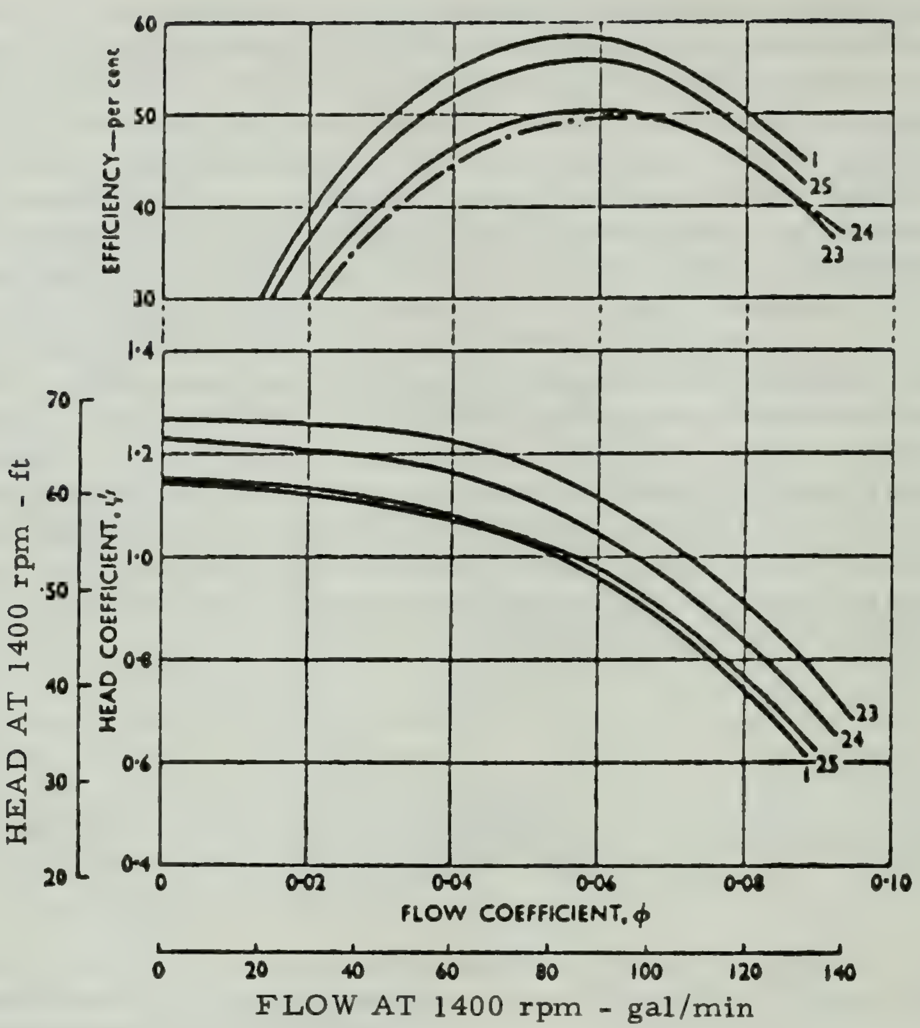
[1]See pages 754 and 755 of reference [52].

The matter of surface roughness has been considered by many authors and the work of Varley [62] may be taken as a guide to its importance. Table 8 and Figs. 23-26 taken from reference [62] show the effects on the performance of a small pump due to coating various surfaces of the impeller with graded sand. The shrouds and vanes of the impellers were independent elements, assembled in such a manner that the net flow areas remained virtually constant for all of the different tests. The figures show clearly that finishing all impeller surfaces is important if maximum efficiency is to be attained. The impeller was double suction, $D = 9.60$ inches, and $D_0 = 3.1$ inches with 5 vanes. Some 16 different vane shapes were investigated and the most efficient was chosen for the tests with the sand coatings. The following quotation from page 966 of reference [62] summarizes the bearing which these tests have on model-prototype relationships: "The base of Fig. 24 being non-dimensional, it is to be expected that similar curves would be obtained from other pumps of like geometric shape, whatever the size, so that the larger the dimensions of the impeller, the less is the advantage of a smooth surface. Conversely, for the scale-model testing of large pumps, the model will require an extreme surface finish to avoid introducing errors due to scale effects."

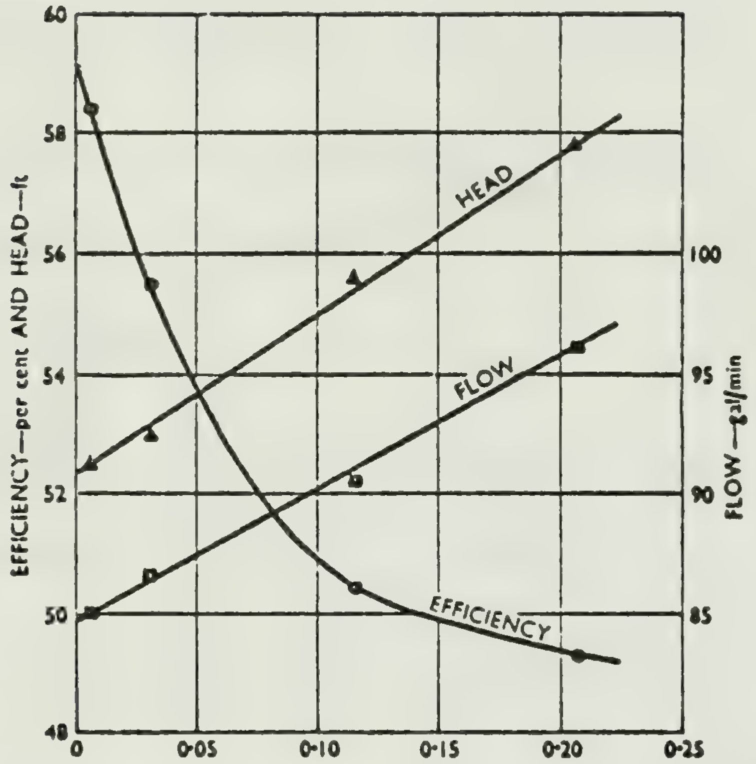
TABLE 8
SAND SIZES FOR ROUGHENED IMPELLERS, VARLEY [62]

Sieve number	Nominal size, in.	Mean nominal size, in.	Mean size by micrometer in.
14 25	0.0474 0.0236	0.0355	0.0341
25 36	0.0236 0.0166	0.0201	0.0191
100 150	0.0060 0.0041	0.00505	0.00513



Impeller number	Surface finish
1	Polished
23	14/25 sand
24	25/36 sand
25	100/150 sand

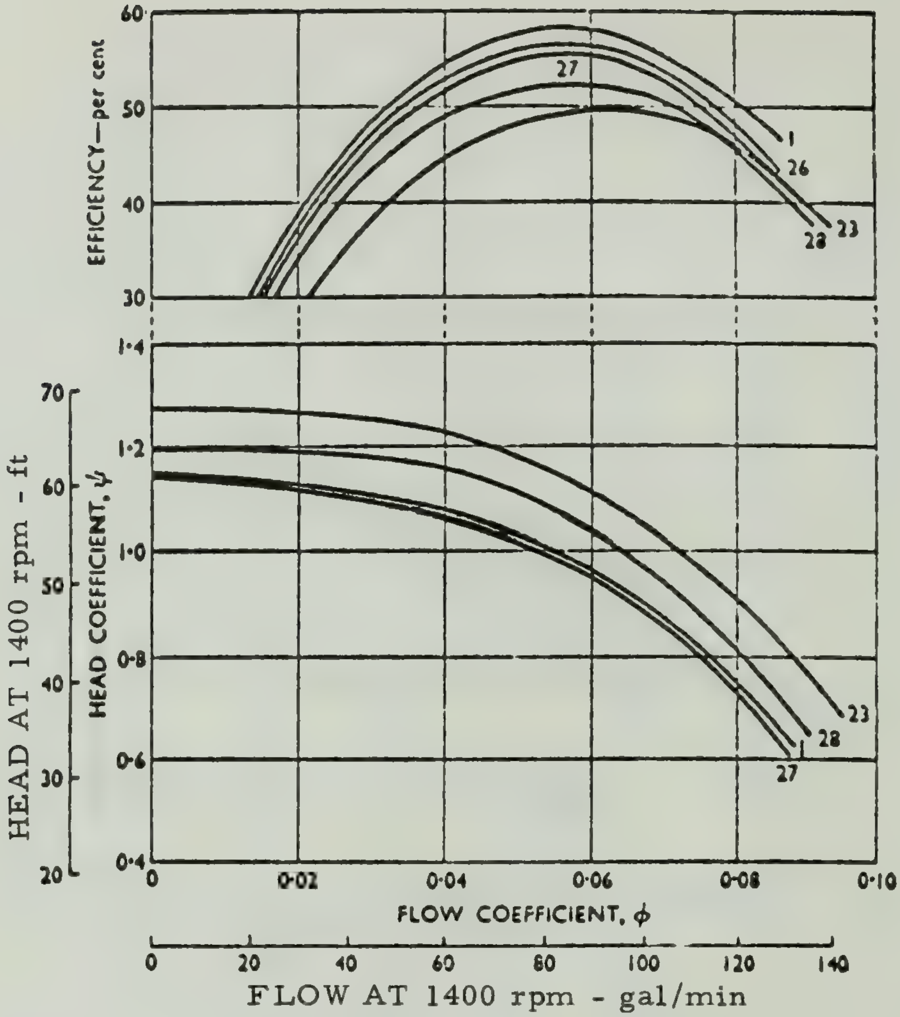
FIG. 23. PUMP CHARACTERISTICS WITH VARYING SURFACE FINISH, VARLEY [62]



ROUGHNESS FACTOR, k/M

n_s = 682 British = 747 U.S.A.

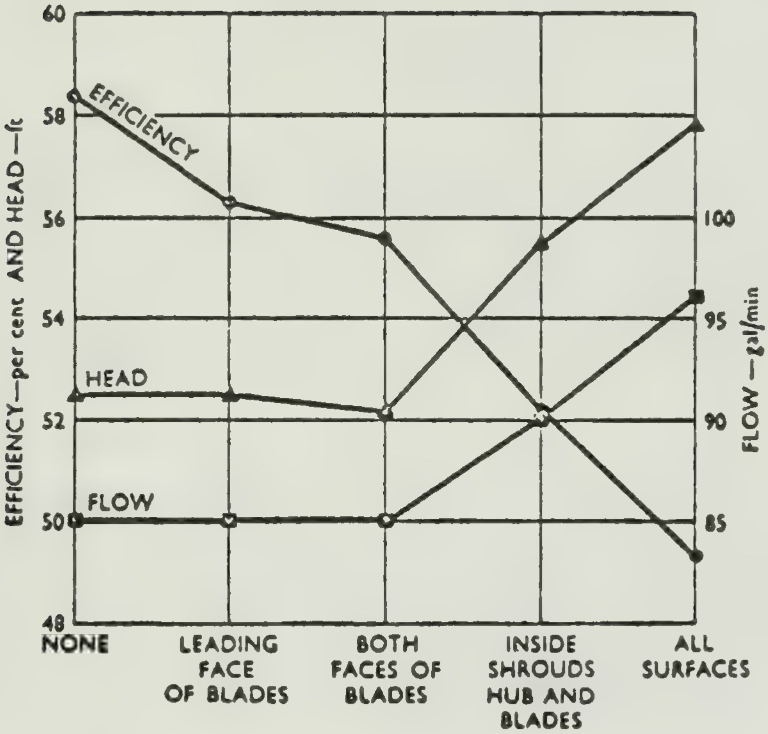
FIG. 24. OPTIMUM PERFORMANCE AT 1400 rpm
WITH VARYING SURFACE FINISH,
VARLEY [62]



The ψ -curve for impeller 26 lies between the curves for impellers 1 and 27.

Impeller number	Components roughened (14/25 sand)
1	None.
26	Leading faces of blades.
27	Both faces of blades.
28	Blades, hub, inside of shrouds.
23	All.

FIG. 25. PUMP CHARACTERISTICS WITH VARIOUS IMPELLER COMPONENTS ROUGHENED, VARLEY [62]



Sand size 14/25 sieve.

FIG. 26. OPTIMUM PERFORMANCE AT 1400 rpm WITH VARIOUS IMPELLER COMPONENTS ROUGHENED, VARLEY [62]

CHAPTER 9

PROPOSED VIBRATION STUDY AND LITERATURE SURVEY

(Austin H. Church)

ABSTRACT

Critical literature survey is made of papers published in English during the past 25 years which pertain to noise and vibration of large pumping units. Probable dynamic systems at Tehachapi are described with possible excitations and causes of vibration reviewed. A set of calculations to investigate vibration problems is recommended. A tentative budget on a time basis for the computational program is given.

A. INTRODUCTION

It is anticipated that none of the vibration calculations outlined in Section C. would be made until the station design is fairly complete, and definite information is available regarding the dynamic properties of the motor, pump, and connecting shaft, as well as the supporting structure. This report is submitted now so that the extent of the vibration study can be considered and incorporated into the overall plans of the pumping facility.

Some 70 pieces of literature in English, dealing with vibration and noise in large pumping units and hydroelectric plants during the past 25 years, were reviewed and are evaluated in the last part of this report. It is interesting to note that a vast majority of these deal with surging in the pipelines rather than with vibration and noise in the rotating members. The computational program of Section C. does not include this surging or water hammer, as this is well covered by others elsewhere. It considers this phenomenon only as a possibly important source of excitation in the vibration of the pumping unit.

When a pump and motor are connected by a shaft the combined system inherently will have three sets of natural frequencies of vibration; one being the transverse, another the torsional, and the third the longitudinal. Each of these should be determined to guard against possible excessive vibration or stress during either steady-state or transient operation. It is not sufficient to consider the natural frequencies of each component part by itself. In fact to be complete and exact, the supporting structure should be incorporated as a part of the motor-shaft-pump combination, particularly for the case of

transverse vibration. Obviously this study must be conducted in one office, such as DMJM, rather than by the individual suppliers of equipment.

The problem of noise generation and its reduction is quite difficult to predict from drawings alone. Perhaps the most efficient manner of handling this aspect is to correct it when it occurs (62C, 57B, 57C, 48A).

Throughout this report specific references to items in the Literature Survey are given in parentheses, such as those just listed. The number refers to the year of the publication and the letter to the particular item of that year as listed in the Survey.

B. DYNAMICS OF SYSTEM

1. Description and Types of Vibration

At this early date the configuration of the Tehachapi Pumping Plant is not fixed. Figure 1 shows two typical schematic plant arrangements which might be used; part A shows a vertical unit, while part B is a horizontal arrangement. The structure supporting the unit may be made of steel, concrete, or a combination of the two (62A, 61A, 46A, 40A). While the pumping unit could be mounted on bedrock, it is usually more desirable to locate it on a structure which would isolate it somewhat from external excitation.

The rotating members of the system are shown in Figure 2 diagrammatically, and this sketch applies equally well to either system of Figure 1. Although it is not shown or designated, there is damping in the pump due to the close clearance of the wearing rings, disk friction, and hydraulic losses; hysteretic damping in the shaft and couplings; and damping in the magnetic fields of the motor and exciter. However, it is difficult to determine the magnitudes of each of these damping factors so that it is frequently necessary to resort to overall values. Response spectra of similar units would be quite helpful in arriving at appropriate approximate coefficients (63A, pp. 215-6).

No bearings, journal or thrust, are shown in Figure 2 since their location is quite speculative at this time. However, their number and location will affect materially the dynamic response of the system.

The systems of Figures 1 and 2 can vibrate in many ways, and each of these should be investigated.

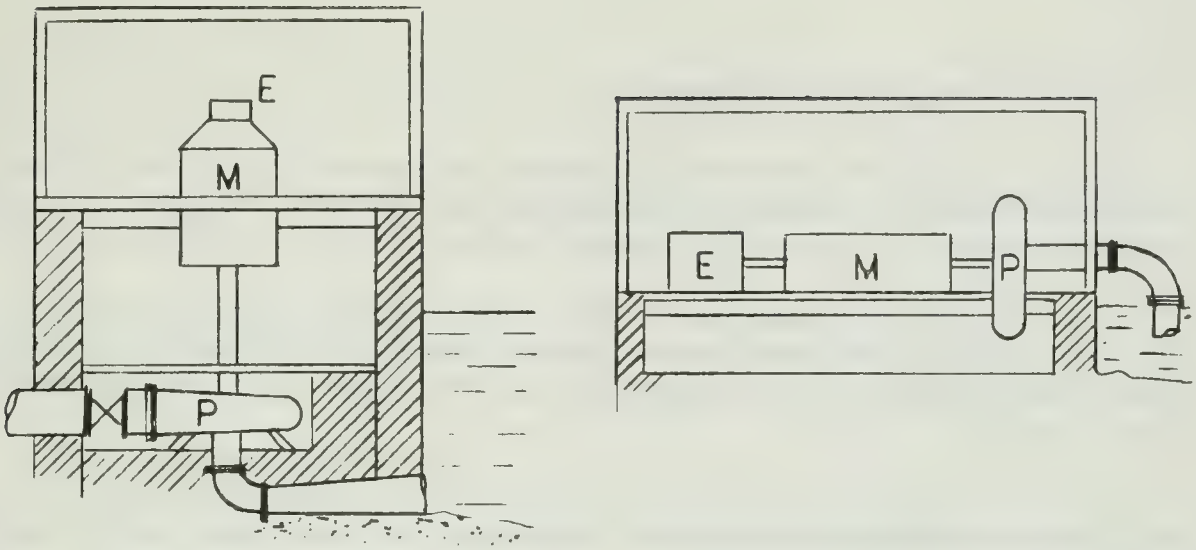


Figure 1

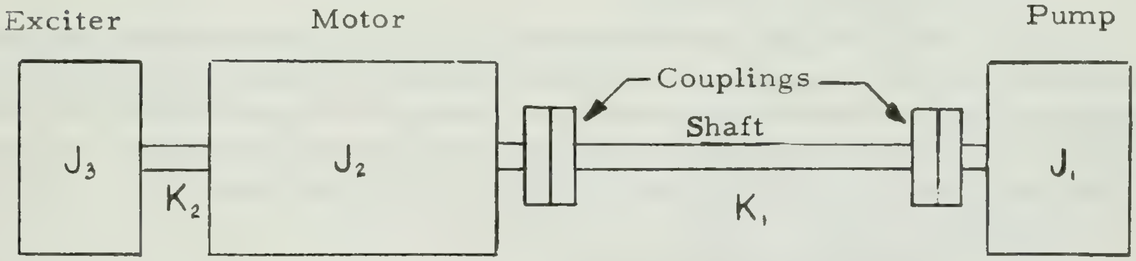


Figure 2

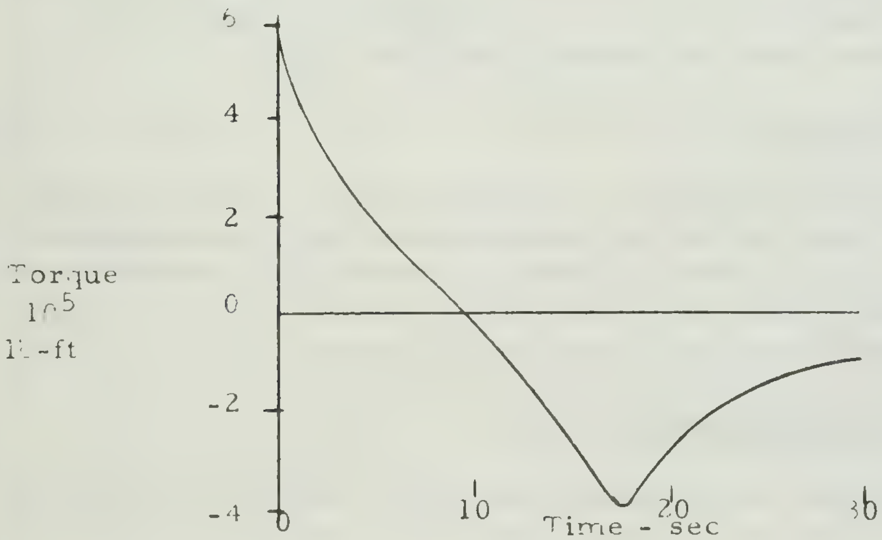


Figure 3

a. Transverse Vibrations

(1) The supporting structure in Figure 1 constitutes either a beam or a frame or portal unit (56B; 60B, pp. 307-18; 63A, pp. 408-13) which has an infinite number of natural frequencies and mode shapes.

(2) The rotating system of Figure 2 will have an infinite number of critical speeds and mode shapes, whose values are influenced by the stiffness of the oil film and bearing support as well as the mass of the bearings (58A, 60D).

(3) Actually items (1) and (2) form a composite system since they interact with each other (60E). However, the analysis of such a composite system can become quite involved, and it may be desirable to use a model (63B) to determine responses.

b. Torsional Vibrations

While the torsional system of Figure 2 actually has an infinite number of natural frequencies and mode shapes, it is generally more convenient and practical to consider the system as lumped into three masses with three natural frequencies (one of which is zero). The value of k_1 on the figure must include the torsional stiffness of the couplings.

c. Longitudinal Vibrations

As noted previously, a thrust bearing is located at some point along the system of Figure 2 to absorb the axial thrust. Longitudinal vibrations can take place about this anchor point (48A, 49A).

2. Causes of Vibration

It is desirable at this point to list some possible causes of vibration in dynamic systems. It is not feasible to refer each item given here to a particular publication in the Literature Survey, as there is considerable duplication and overlap. However, the following general references are helpful: 61A; 56E; 61H; 63A, pp. 111-2.

a. Unbalance in Rotating Members

(1) This may increase with time due to chemical deposits on the pump vanes.

(2) It may increase with the speed as in the case of somewhat loose armature windings.

(3) The rotor may be flexible (rather than stiff) so that the amount of unbalance changes with the speed (61G).

b. Oil Whip in Bearings

Creates a disturbing force at 1/2 the rotative speed.

c. Mechanical Failure

This includes rubbing of parts, loose parts, and excessive wear. Frequency is twice the rotating speed.

d. Worn Bearings or Wearing Rings

Reduces the damping and so increases the amplitude. For extreme wear, the bearing may lose constraint of the system at that point to alter the natural frequencies drastically.

e. Interference or Defective Teeth of Gears

Frequency equals number of teeth that are defective times the rpm of gear. This is not applicable to Tehachapi, but is considered for general completeness.

f. External Excitation

(1) Transmitted forces or motions from adjacent units could cause a beating action in the unit or the structure.

(2) Transmitted from the suction or discharge pipe due to water hammer (53B; 55D). Frequency could be Fourier series harmonics of the more or less square wave main excitation.

g. Misalignments and Distortions of the System

(1) During assembly of couplings or casings. Tightening bolts unevenly, etc.

(2) Thermal stresses in unit. During machining the crystalline

structure of a metal may be damaged locally to change the rate of heat transfer, and cause uneven expansions.

(3) Thermal stresses in foundation. Portions of foundation subjected to local heating should be insulated to prevent distortions of the frame.

h. Local Resonance of a Part or Component

i. Bent Shaft

Initially, or due to stress in operation.

j. Bearing Defects

Out-of-roundness, improper clearances, surface defects.

k. Magnetic Effects

l. Hydrodynamic or Hydrostatic Forces in Unit

Pressure pulsations in hydraulic machinery (54A; 56A, pp. 7-9).

m. Magnetic Pulsation in Motor

n. Torque Pulsations of Motor Driven as Generator

Transient torque = $A\ e^{-Bt} \sin \omega t - C\ e^{-Dt} \sin 2\omega t + E\ e^{-Ft}$
where: A, B, ... , F arc constants for a particular motor.

o. Cavitation in Pump (61D)

p. Nonuniform Shaft Cross Section

Keyway or spline shaft may cause variations in natural frequency, and hence vary the vibration response for given rotative speed.

Obviously, the effects of these causes are magnified as the exciting frequency approaches a natural frequency of the system, or some part of the system.

3. Vibration Problems at Tehachapi

The rather extensive list of Section B. 2. needs to be reviewed as it applies to the Tehachapi Project. Two types of vibration can occur - the steady-state and the transient.

a. Steady-state Vibrations

When the unit is operating normally, it is necessary to insure that the magnitude of the excitations is held to a minimum, and that their frequencies do not approach the natural frequencies of the dynamic system. The predominating items in the list of Section B. 2. for the various types of vibration mentioned in Section B. 1. are:

(1) Transverse: a., b., c., d., f., g., h., i., j., k., l., m., o., p.

(2) Torsional: b., c., d., f., g., h., j., k., l., m., n., o.

(3) Longitudinal: c., d., f., g., h., i., j., k., l., m., o.

b. Transient Vibrations

Transient situations are vitally affected by the magnitude of the surges in the pipe lines (39A, 53B, 55D). In addition to the normal procedures of starting up and shutting down the unit, the following list of possible transients is taken from Ref. 39A:

(1) Power Failure Due to an Electrical Fault

This condition is certain to occur more or less frequently. The motor circuit breaker is tripped open and the pump discharge valve automatically starts to close.

(2) Delay in Tripping Motor Breaker

If a motor is not immediately disconnected from a short circuit, it will act for a short time as a generator, giving additional retarding torque and resulting in more rapid deceleration than normal.

(3) Failure of Valve to Operate When Power is Shut Off

In this case, the flow in the pipe line will reverse; the pump and motor will reverse and attain runaway speed in the reverse direction.

(4) Delay in Action of the Valve

If the operation of the valve, in case of power failure, is delayed until full reverse flow is established, and the valve then closes rapidly, severe water hammer will result.

(5) Failure of Shaft Coupling

The flywheel effect of the motor is removed, the speed will drop rapidly, and the pump will reverse rotation.

(6) Seal-ring Seizure

The speed will drop rapidly, but the pump will not reverse.

Calculations to predict potential difficulties for both the steady-state and the transient cases will be outlined in the next section.

C. COMPUTATIONAL PROGRAM

It is rather difficult to outline a computational program for an elastic system that is not even conceived, but this will be done on the basis that Figures 1 and 2 apply. It will follow the outline given in Section B. 3. It will be assumed that the transverse natural frequencies, or critical speeds, of the pump and the motor by themselves have been determined by the supplier and found to be well removed from the excitations listed in Section B. 2.

1. Steady-state Vibrations

The magnitude and location of the force or motion excitation for steady-state conditions is usually unknown. However, an excitation applied at or near a natural frequency anywhere along the shaft, except at a node, will produce a large response or resonance. Also, the pump and motor have relatively unknown amounts of damping. The effect of the damping is to lower the natural frequencies somewhat below those of the same system undamped.

In view of these uncertainties, the usual approach is to determine the

undamped natural frequencies and keep the exciting frequencies well away from them, say 20%. This procedure will be followed for the combined system shown on Figure 2 with the bearings located in place.

a. Transverse Vibrations

(1) Assume that the unit of Figure 2 is mounted on a rigid baseplate and determine the first undamped natural transverse frequency by the Prohl method (45A) including the gyroscopic effect of the impeller and armature.

(2) Reduce the supporting structure of Figure 1 to that of a portal frame (depending on the design), or to a beam with either simple or fixed ends (again, according to the design), and obtain the deflection and slope response at the motor (or motor and pump combination) to a unit load applied at the rotative speed of the unit. This needs to be done twice - once for a unit force, and then again for a unit moment or torque applied at the motor or motor-pump combination. The first three or four natural frequencies of the structure and responses need be found (56B, 60E).

(3) From items (1) and (2) reasonable displacement impedance curves can be drawn to obtain approximate resonances of the unit-structure combination. These natural frequencies should be about 20% away from the exciting frequencies or integer multiples of them.

(4) Examine the design in general looking for local resonances and considering the excitations listed in Section B. 2.

b. Torsional Vibrations

(1) Use Holzer tables (46C; 63A, pp. 303-24) and/or displacement mobility methods to determine the three torsional undamped natural frequencies of the system of Figure 2.

(2) The values found in (1) should be checked against the excitation frequencies listed in Section B. 2. For instance, the number of impeller vanes times the rpm, and so on.

c. Longitudinal Vibrations

Reduce the system of Figure 2 to that of a longitudinal bar supported at the thrust bearing. Calculate the longitudinal natural frequencies

(48A; 49A) of the bar. Probably these will be quite high compared to the exciting frequencies. If not, the method of item a. (3) of this section could be used, i. e., match impedance curves.

2. Transient Vibrations

The range of speed of the unit undergoing the transient excitation outlined in Section B. 3. is generally less than that of the steady-state speed, so that it is hoped that the natural frequencies found in Section B. 1. will be above the steady-state speeds to avoid going through resonances. However, the curves of Refs. 53B, 55D and 39A show that the transient speed of the unit fluctuates rather rapidly so that resonant conditions are not apt to have sufficient time to build up. The transverse and longitudinal vibrations during the transients listed are generally of a secondary nature, and it is felt that they can be ignored. Hence the focus of the investigation here should be to avoid excessive torsional stresses in the rotating members.

The transient cases outlined in Section B. 3. will now be considered from the torsional viewpoint.

a. Power Failure due to an Electrical Fault

This situation is well outlined in Refs. 39A, 53B, and 55D, of which 55D is the most definite, although they all show similar results. The following discussion refers particularly to Ref. 55D.

Figure 2 of that paper is based upon the pump characteristic diagram and the water-hammer characteristics of the pipelines. If this data is traced back through the characteristic diagram of the pump (for which it is not easy to get accurate results because of the small scales given in the published paper) we find that the pump torque-time curve is similar to that shown in Figure 3 of this report. During this transient the motor is being driven as a generator so it has a retarding torque given as item n on page 9-6 of this report. This acts in the same direction as the retarding pump torque during the first 10 seconds, and then in the opposite direction for the remainder of the transient.

The exact pump torque-time curve (similar to Figure 3 of this report) would be evolved from the water-hammer study for a particular design, and the equation for the transient torque-time relationship for the motor being driven as a generator could be obtained from the motor supplier.

With this data it would be possible to determine the angle of shaft twist (and hence the shaft stress) as a function of time using phase-plane techniques or Laplace transforms, either with or without superposition (depending on which technique is simplest and most rapid). It would be an interesting problem, but not simple. It would be necessary, of course, to include the initial shaft deflection and the initial angular velocity of the shaft in the calculation.

These calculations could be made including the effect of the damping in the system. However, this effect would probably be small, and it would be conservative (i.e., give higher stresses) to ignore the damping. It would also reduce the complexity of the calculations by about an order of magnitude. Hence it is proposed to ignore the effect of damping here.

b. Delay in Tripping Motor Breaker

The solution for this case would follow that for case a. It would require the transient curves (similar to Figure 2 of Ref. 55D) with the pump torque-time curve included.

c. Failure of Valve to Operate when Power is Shut Off

Same comments as for case b.

d. Delay in Action of the Valve

Same comments as for case b.

e. Failure of Shaft Coupling

No vibration would result. The motion of both the pump and motor could be obtained from $T = J\alpha$, where T is obtained from the water-hammer study, and from the motor characteristics when being driven as a generator.

f. Seal-ring Seizure

It is assumed that this means that the impeller stops instantaneously, and that the motor armature vibrates as though it had a single degree of freedom. If the motor angular velocity and torque is known at the instant when the impeller stops rotating, it would be relatively simple to find the maximum resulting shaft stress due to the accident.

D. LITERATURE SURVEY

1931

A "CENTRIFUGAL PUMPS OPERATED UNDER ABNORMAL CONDITIONS" by C. P. KITTREDGE with D. THOMA, Power, June 2, 1931, pp. 881-4.

Brief account of pioneering work on the characteristics of a centrifugal pump acting as a turbine.

1937

A "COMPLETE CHARACTERISTICS OF CENTRIFUGAL PUMPS AND THEIR USE IN THE PREDICTION OF TRANSIENT BEHAVIOR" by R. T. KNAPP, ASME Trans., Nov. 1937, pp. 683-9. See also pp. 676-80 of ASME Trans., Nov. 1938, for discussion.

First plot of complete characteristic diagram of a pump, based on a suggestion by von Karman. These diagrams are essential to obtain the performance of a turbomachine under transient conditions.

1939

A "TYPICAL ANALYSIS OF WATER HAMMER IN A PUMPING PLANT OF THE COLORADO RIVER AQUEDUCT" by R. M. PEABODY, ASME Trans., Feb. 1939, pp. 117-24.

Uses method of arithmetic integration of a Karman-Knapp characteristic diagram of a pump to obtain head, speed, and flow versus time curves for various types of transient conditions. He includes the characteristics of the valve in these studies. While the curves are derived primarily for water-hammer studies, they are very interesting from the viewpoint of transient vibrations of the pumping unit.

B "THEORY OF RESONANCE IN PRESSURE CONDUITS" by C. Jaeger, ASME Trans., Feb. 1939, pp. 109-15.

Interesting discussion of the possibility that the period of water-hammer surges may coincide with the natural period of the pipe to cause resonance, with a destructive effect on the pump.

1940

A "EXPERIMENTAL AND THEORETICAL INVESTIGATION OF A TURBINE FOUNDATION" by S. VESSELOWSKY, ASME-JAM, June 1940, pp. A-63 to A-70; Discussion JAM,Sept. 1941, pp. A-141 to A-142.

Analytical study of turbogenerator mounted on heavy, baseplate supported on steel work using Lagrange's equations of motion to obtain natural frequencies and mode shapes. Verified experimentally with good agreement.

B "TORSIONAL VIBRATION IN GEARED-TURBINE PROPULSION EQUIPMENT" by H. PORITSKY & C. S. L. ROBINSON, ASME-JAM, Sept. 1940, pp. A-117 to A-124.

Good discussion of damping in torsional systems. Application is to determine stresses in marine propulsion systems.

1942

A "SOME PROBLEMS IN THE SELECTION AND OPERATION OF CENTRIFUGAL FOR OIL AND GASOLINE PIPE LINES" by A. HOLLANDER, Trans. ASME, Aug. 1942, pp. 607-17.

This is a "state of the art" paper. Transient conditions or vibration problems not considered. Primarily concerned with viscosity effects.

1944

A "DYNAMIC EFFECTS" by F. M. LEWIS, Chapter II of "Marine Engineering", Edited by H. L. Seward, Soc. Nav. Arch. & Mar. Engrs., 1944, pp. 76-140.

Good review of torsional vibration theory with extensive bibliography (up to 1944). Good information on equivalent torsional systems.

1945

A "A GENERAL METHOD FOR CALCULATING CRITICAL SPEEDS OF FLEXIBLE ROTORS" by M. A. PROHL, ASME-JAM, Sept. 1945, pp. A-142 to A-148.

An excellent tabular method (similar to Holzer tables) to determine all critical speeds of stepped rotors having any number of bearings, and can include the gyroscopic effects of the rotors. Amenable to programming on computer. Widely used in industry.

1946

A "DYNAMIC FACTORS AFFECTING TURBINE FOUNDATION DESIGN" by S. VESSELOWSKY, Electrical World, Aug. 3, 1946, pp. 60-1.

Lists precautions in designing foundations such as avoiding overhung loads, thermal stresses (insulate parts subject to heat). Recommends keeping configuration simple for ease of analysis, and providing simple ways to alter it to change the natural frequencies if necessary.

B "CALCULATION OF MULTIPLE-SPAN CRITICAL SPEEDS OF FLEXIBLE SHAFTS BY MEANS OF PUNCHED-CARD MACHINES" by A. W. RANKIN, ASME-JAM, 1946, pp. A-117.

He first transforms shaft into an equivalent circuit, which is then solved by a computer. He uses admittances and impedances. Seems unnecessarily awkward, unless you are an E. E. trying to get along in a mechanical job.

C "FORCED TORSIONAL VIBRATIONS WITH DAMPING" by J. P. DEN HARTOG and J. P. LI, ASME-JAM, Dec. 1946, p. A-72.

Interesting paper adapting Holzer table method to damped, lumped systems subjected to external excitations.

1947

A "DESIGN OF VIBRATION-ISOLATING BASES FOR MACHINERY" by C. E. CREDE and J. P. WALSH, ASME-JAM, March 1942, pp. A-7 to A-14.

Excellent paper on the general subject of vibration isolation. Theory verified experimentally.

1948

A "VIBRATION AND SOUND" by P. M. MORSE, McGraw-Hill, 1948.

One of the modern classics on acoustics.

1949

A "LONGITUDINAL VIBRATIONS OF MARINE PROPULSION-SHAFTING SYSTEMS" by J. R. KANE and R. T. McGOLDRICK, Trans. S. N. A. & M. E., Nov. 1949.

Could be applied to axial forces in the motor-pump unit. Body of paper is mostly descriptive, but appendices have helpful material. Uses mobility and impedance methods.

B "ELEMENTS OF GRAPHICAL SOLUTION OF WATER-HAMMER PROBLEMS IN CENTRIFUGAL PUMP SYSTEMS" by A. J. STEPANOFF, ASME Trans, July 1949, pp. 515-34.

Good general review of elementary water-hammer analysis, but nothing new. Cites cases of pump damage due to water hammer - bent shafts, cracked pump casings, etc.

C "DYNAMIC PRINCIPLES OF MACHINE FOUNDATIONS AND THE GROUND" by J. H. A. CROCKETT and R. E. R. HAMMOND, Proc. I. M. E., v. 160, 1949, pp. 512-31.

Discussion of the effect of soil mechanics on the effectiveness of isolation. Done for one and two degrees of freedom systems. Considers the isolation of stress waves travelling through the ground by use of cork bases. Interesting paper.

D "HOLZER METHOD FOR FORCED-DAMPED TORSIONAL VIBRATIONS" by T. W. SPAETGENS, ASME-JAM, March 1949, p. A-12.

Not too helpful. Topic is covered better in Refs. 46C, or 63A, pp. 310-24.

1950

A "DEVELOPMENT OF THE HYDRAULIC DESIGN FOR THE GRAND COULEE PUMPS" by C. BLOM, ASME Trans., 1950, pp. 53-70.

An interesting discussion of hydraulic pump design, but no information given regarding vibration or transient response.

1951

A "PRINCIPLES OF FOUNDATION DESIGN FOR ENGINES AND COMPRESSORS" by W. K. NEWCOMB, ASME Trans., April 1951, pp. 307-18.

As title implies this paper applies to reciprocating machines. Discusses some soil mechanics and mat designs to support machinery bases. Rather empirical in parts. Good, but not particularly applicable to Tehachapi.

B "CONTRIBUTIONS TO IMPROVED ACCURACY IN THE CALCULATION AND MEASUREMENT OF TORSIONAL VIBRATION STRESSES IN MARINE PROPELLER SHAFTING" by S. ARCHER, Proc. I. M. E., 1951, pp. 351-66.
"TORSIONAL VIBRATION DAMPING COEFFICIENTS FOR MARINE PROPELLERS" by S. ARCHER, Engineering, 1955, p. 594.

This pair of papers is devoted primarily to determining the torsional damping constant of a ship propeller. He lists values by various authorities which range from 28.6 to 39.2 lb-sec-ft/rad., with an average at about 36. It would be desirable to have a similar set of figures for a pump impeller in its casing.

1952

A "HOW TO CUT VIBRATIONS IN BIG TURBINE-GENERATOR FOUNDATIONS" by F. P. WITMER, Power, Nov. 1952, pp. 86-7.

General discussion of steel versus concrete foundations. Steel gives more available space and is light, but is subject to local resonances. Concrete has mass, but is hard to alter. The combination of steel and concrete is good, but expensive.

B "MEASUREMENT OF HYDRAULIC-TURBINE RUNNER VIBRATION" by J. PARMAKIAN and R. S. JACOBSEN, ASME Trans., July 1952, pp. 733-41.

Measurement of turbine runner vibration using SR-4 strain gages in a pressure cell. Signal taken off through slip rings. Vibration cured by grinding down trailing edge and modifying runner shape.

1953

A "COMPLETE CHARACTERISTIC CIRCLE DIAGRAMS FOR TURBO-MACHINERY" by W. M. SWANSON, Trans. ASME, July 1953, pp. 819-26.

Gives the pump characteristic diagrams, based on test results for two types of pumps: axial flow with n_s = 13,500 and mixed flow with n_s = 7550. These can be compared with the results for the pump of Ref. 37A, which has a specific speed of 1270. The curves are given on a percentage basis for greater usefulness.

B "PUMP SURGES AT LARGE PUMP INSTALLATIONS" by J. PARMAKIAN, ASME Trans., Aug. 1953, pp. 995-1006.

Gives analysis, using characteristic diagram and water-hammer techniques, of transient performance of large pump due to power failure. The characteristic diagram is derived from the usual normal pump performance curves. Very interesting and useful paper. See also Ref. 55D.

1954

A "VIBRATION OF THE GRAND COULEE PUMP-DISCHARGE LINES" by J. PARMAKIAN, ASME, July 1954, pp. 783-90.

One unit produced severe steady-state vibration in exposed pipe line at a frequency of 23 1/3 cps, which was traced to the pump. The pipe shell was stiffened by adding circumferential rings to raise natural frequency. The magnitude of the pulses in the pump were reduced by increasing the radial clearance between the impeller and diffuser vanes. Also by removing the main splitter vane opposite the volute tongue. This increased the flow and efficiency at lower heads.

B "GRAND COULEE MODEL-PUMP INVESTIGATION OF TRANSIENT PRESSURES AND METHODS FOR THEIR REDUCTION" by E. LINDROS, ASME Trans., July 1954, pp. 775-82.

Chiefly a discussion of the instruments and their calibration used in studying the problem of Ref. 54A on a model of the pump.

C "THE ANALYSIS AND SYNTHESIS OF VIBRATION SYSTEMS" by R. E. D. BISHOP, J. Royal Aero. Society, Oct. 1954, pp. 703-19.

Development of receptance method of determining the dynamic response of lumped systems.

1955

A "THE ANALYSIS OF VIBRATING SYSTEMS WHICH EMBODY BEAMS IN FLEXURE" by R. E. D. BISHOP, Proc. I. M. E. v. 169, 1955, pp. 1031-50.

Includes tables, based on the receptance method, to determine steady-stage response of beams of constant cross section with all types of end conditions. Excitation is force or moment applied at any point along span. Slope or deflection response at any point along beam can be obtained with relatively simple calculation. Excellent and very useful paper.

B "SELECTION OF HIGH-LIFT PUMP MOTORS AND CONTROLS" by A. C. MICHAEL and G. I. STORMONT, J. A. W. W. A., 1955, pp. 39-48

High-lift turns out to be about 80 feet. Whole paper is elementary and trivial.

C "PREDICTION AND MEASUREMENT OF VIBRATION IN MARINE GEARED-SHAFT SYSTEMS" by H. G. YATES, Proc. I. M. E., v. 169, 1955, pp. 611-42.

Responses are calculated using admittance and impedance based on mass-inductance analogy diagrams. He includes the effect of damping (resistance) and includes a discussion on this using the Q factor.

D "PRESSURE SURGES IN PUMP INSTALLATIONS" by J. PARMAKIAN, Paper No. 2760, Trans. ASCE, 1955, pp. 697-720.

This paper is identical in coverage to Ref. 53B, except that the data and curves apply only to the Tracy Pumping Plant, rather than three plants. Thus the figures and results are more useful and specific.

E "EXTENDED THEORY OF THE VISCOUS VIBRATION DAMPER" by F. M. LEWIS, ASME-JAM, Sept. 1955, pp. 377-82.

Demonstrates that the fixed-point theorem applies to multimass systems, and can be used to obtain the optimum damping factor. Very good analysis, and useful paper.

1956

A "MECHANICAL VIBRATIONS" by J. P. DEN HARTOG, 4th Edition, McGraw-Hill, 1956.

A general text with many practical applications obviously taken from his extensive consulting experience.

B "THE VIBRATION OF FRAMES" by R. E. D. BISHOP, Proc. I. M. E., v. 170, 1956, pp. 955-68.

Excellent development for determining the natural frequencies of structures consisting of beams and columns, using receptance methods. Extension of Ref. 55A to structures.

C "FIELD TESTING A REVERSIBLE PUMP-TURBINE" by F. E. JASKI, Mechanical Engineering, Feb. 1956, pp. 141-4.

Paper largely descriptive of operation and hydraulic testing of the unit.

D "A SURVEY OF AERODYNAMIC EXCITATION PROBLEMS IN TURBOMACHINES" by A. SABATIUK and F. SISTO, Trans. ASME, April 1956, pp. 555-64.

Discussion of rotating stall and flutter in compressors and steam or gas turbines. Aerodynamics, not hydrodynamics.

E "VIBRATION IN VERTICAL PUMP SYSTEMS" by J. H. McKENDREE and L. A. MARCH, Trans. AIEE, Part III, Dec. 1956, pp. 1169-77.

Very good discussion of causes of vibration in vertical centrifugal pumps requiring more than 75 horsepower; including a check list of suggested causes and their possible cures.

F "HYDRAULIC TURBINE RUNNER VIBRATION" by R. M. DONALDSON, ASME Trans., July 1956, pp. 1141-7.

Laboratory investigation of runner vibrations as related to profile of exit edge. The results of these tests were related to field experiences.

G "HYDRAULIC TRANSIENTS IN CENTRIFUGAL PUMP SYSTEMS" by C. P. KITTREDGE, ASME Trans., Aug. 1956, pp. 1307-22.

The pump characteristic diagram is plotted on a dimensionless basis, and computation method applied to obtain the transient pump performance plotted against time.

H "PRACTICAL SOLUTION OF TORSIONAL VIBRATION PROBLEMS" by W. KER WILSON, 3rd Edition, John Wiley, 1956.

Encyclopaedic discussion of torsional vibration problems. Particularly good on reducing actual system to an equivalent one.

I "SUPPRESSION OF PUMP VIBRATIONS SET UP AT STARTING UP - PREOPENING METHOD" by F. NUMACHI, ASME Trans., Nov. 1956, pp. 1735-40.

Uses the characteristic diagram to determine the amount of "preopening" or by-passing around the discharge valve needed to reduce the magnitude of the surge at starting. Good paper.

1957

A "VIBRATION IN HYDROELECTRIC INSTALLATION" by S. LELIAVSKY, Water Power, April 1957, pp. 132-7.

This is largely a review of elementary vibration theory. Nothing significant.

B "LICK PUMP VIBRATION" by D. NOBLES and J. R. HAMILL, Industrial Power, Feb. 1957, pp. 16, 17, 44, 45.

Description of attempts to prevent noise that appeared to originate in basement pumps from being transmitted to an office through the discharge pipe. Actually it traveled up the building steel work.

C "HANDBOOK OF NOISE CONTROL", Edited by C. M. HARRIS, McGraw-Hill, 1957.

This is an excellent book, written by experts in the field of acoustics. Covers all phases of the subject.

1958

A "PREDICTION OF CRITICAL SPEEDS OF STEAM TURBINES BY THE DYNAMIC STIFFNESS METHOD" by W. J. CARUSO, Collequium on Mechanical Impedance Methods, Edited by R. Plunkett, ASME 1958, pp. 137-45.

Very interesting paper showing the effect of the flexibility of the supports in altering the critical speeds of rotating machinery. Reflects General Electric practice, but the results are general.

B "CONTROL OF VIBRATION AND NOISE FROM CENTRIFUGAL PUMPS" by L. M. EVANS, Noise Control, Jan. 1958, pp. 28-31.

Discussion applied to rather small pumps. Following suggestions are made: mount pump on isolator pads; enclose it in an acoustic box; use acoustic tile in the pump room; use relatively large clearance between impeller and tongue of volute; use rubber hoses on suction and discharge lines with hose axis perpendicular to pump axis; journal bearings rather than ball or roller bearings, etc.

1959

A "USE VIBRATION ANALYSIS TO PINPOINT TROUBLE" by L. N. DAWSON, Petroleum Engineer, April 1959, pp. C24-30.

Discussion of probable sources of trouble with vibration in small centrifugal pumps and how to locate them with a portable electronic

vibration analyzer. Stresses the importance of good balancing and alignment for long life.

B "HOW TO MUFFLE NOISY PUMPS" by L. M. EVANS, Mill and Factory, April 1959, pp. 118-9.

This says nothing!

1960

A "CHARACTERISTICS OF CENTRIFUGAL PUMPS AND COMPRESSORS WHICH AFFECT THE MOTOR DRIVER UNDER TRANSIENT CONDITIONS" by H. A. WIEGAND and L. B. EDDY, Trans, AIEE, July 1960, pp. 150-6.

Primarily a description of small or medium sized pump and compressor characteristic curves for various specific speeds with corresponding torque-capacity curves. A bit on mass moment of inertia, but nothing significant. Chiefly to acquaint electrical engineers with the torque requirements of pumps and compressors.

B "THE MECHANICS OF VIBRATION" by R. E. D. BISHOP and D. C. JOHNSON, Cambridge University Press, 1960.

An excellent general text on vibration. Emphasis is on receptance method throughout.

C "SHOCK AND VIBRATION HANDBOOK" Edited by C. M. HARRIS and C. E. CREDE, 3 volumes, McGraw-Hill, 1961.

This is an excellent set of books, written by outstanding experts in the field. Covers most phases of the field in an up-to-date manner.

D "SIMPLIFIED VIBRATION ANALYSIS" by A. H. CHURCH, Machine Design, May 26, 1960, pp. 135-42.

Discusses effect of stiffness of bearing support and oil film as well as the mass of the bearing on the natural frequencies of beams or shafts.

E "INTERIOR RECEPTANCE OF BEAMS" by R. E. BISHOP, J. Mech. Engrg. Sciences, vol. 2, Nov. 1, 1960, pp. 1-15.

Extension of Ref. 55A to facilitate the calculation of the receptances along beams.

1961

A "VIBRATION OF VERTICAL PUMPS" by A.KOVATS, ASME Paper No. 61-HYD-10, 1961.

Rather poorly written paper on medium-size submerged pumps, with rather elementary generalized suggestions for analysis for vibration.

B "VIBRATION PROBLEMS IN HYDRAULIC STRUCTURES" by F. B. CAMPBELL, J.ASCE, HY 2, March 1961, pp. 61-77.

This is a sketchy review of elementary vibration principles. Rather trivial.

C "ANALYSIS FOR CALCULATING LATERAL VIBRATION CHARACTERISTICS OF ROTATING SYSTEMS WITH ANY NUMBER OF FLEXIBLE SUPPORTS"
"Part 1 - THE METHOD OF ANALYSIS" by E. C. KOENIG, ASME-JAM, Dec. 1961, pp. 585-90.
"Part 2 - APPLICATION OF THE METHOD OF ANALYSIS" by T. G. GUENTHER and D. C. LOVEJOY, ASME-JAM, Dec. 1961, pp. 591-600.

This pair of papers, written by engineers at Allis-Chalmers, presents a method for calculating the transverse vibrations of rotating shafts having multiple supports. It includes elasticity and damping in the bearings. The method uses matrix algebra, and is programmed for solution on the IBM 704 computer. Several interesting curves, showing the effect of support flexibility, are given in Part 2. If a computer is to be used in analyzing the vibration conditions at Tehachapi, this computer program could be very helpful.

D "REDUCTION OF TURBINE RUNNER VIBRATION AND NOISE" by F. C. TAYLOR, ASME J. Engrg. for Power, April 1961, pp. 184-9.

This is one of a number of papers on runner vibration. Three general approaches are used to mitigate the effects of the von Karman vortices at the trailing edge. One is to tie the runners together with struts, rings or wedges to raise the natural frequency. Another is to

introduce air bubbles to absorb the energy of vibration. Third is to alter the shape or thickness of the trailing edge to alter the vortex frequency.

E "DESIGN OF A COMPUTER PROGRAM TO DETERMINE THE NATURAL FREQUENCIES AND NORMAL MODES OF VIBRATION OF IN-LINE MECHANICAL SYSTEM OF SPRINGS AND MASSES" by T. O'CALLAGHAN, Soc. Indust. & Applied Math. J., June 1961, pp. 294-310.

This is a mathematical paper that is hard to follow or apply to a practical problem.

F "COMPLETE PUMP CHARACTERISTICS AND THE EFFECTS OF SPECIFIC SPEEDS ON HYDRAULIC TRANSIENTS" by B. DONSKY, ASME J. of Basic Engrg., Dec. 1961, pp. 685-99.

This is similar to Ref. 53A, but diagrams are larger and more complete. Diagrams drawn for specific speeds of 1800, 7600 and 13,500.

G "BALANCING FLEXIBLE ROTORS" by A. H. CHURCH and R. PLUNKETT, ASME Trans. J. of Engrg. for Ind., Nov. 1961, pp. 383-9.

Discusses method of balancing flexible rotors using mobility and impedance method.

1962

A "FOUNDATIONS FOR HIGH-SPEED MACHINERY" by J. S. SOHRE, ASME Paper No. 62-WA-250, 1962.

An excellent discussion of foundation design. It reflects intelligent American practical experience as well as European practice as shown by the list of references. While it is primarily directed to steam turbine-generator applications, the principles outlined apply equally well to a large pumping unit, or station.

B "METHODS OF NOISE AND VIBRATION SUPPRESSION" by C. T. MOLLOY, ASME Paper No. 62-MD-14, 1962.

Author applies mobility and impedance methods by drawing equivalent circuits on a mass-inductance basis to the familiar acoustics path: source-transmission-radiator. He discusses in general terms the insertion of filters into the transmission portion. While interesting, it does not have too much direct application to Tehachapi.

C "NOISE REDUCTION THEORY APPLIED TO A LARGE POWER PLANT" by J. PARMAKIAN, Proc. ASCE, PO1, May 1962, pp. 19-37.

As stated in title, this is application of rather standard methods of reducing the noise level in a hydroelectric power plant. Turbine is encased in an acoustic enclosure, and the control room paneled to lower noise level.

D "VIBRATION TRANSMISSION AND IMPEDANCE OF BASIC FOUNDATION STRUCTURES" by D. V. WRIGHT, D. F. MILLER, J. G. AKEY and A. C. HAGG. Westinghouse Research Laboratories Report No. 62-917-515-RI for Bureau of Ships, Oct. 1962, 260 pages.

This is an excellent comprehensive report on the vibration in portal frames using impedance methods combined with matrix algebra for programming on an IBM 7090 computer. Also discusses the use of plastic models.

1963

A "MECHANICAL VIBRATIONS" by A. H. CHURCH, 2nd Ed., Wiley, 1964.

A general text on the subject.

B "VIBRATION CHARACTERISTICS OF PROPULSION MACHINERY SUBBASES AS DETERMINED BY PLASTIC-MODEL TESTS" by G. O. SANKEY and D. V. WRIGHT, Westinghouse Research Laboratories Research Report No. 63-917-515-R3, Nov. 1963.

For involved structures, plastic models have advantages in determining response spectra. They can be fabricated, tested and modified at relatively low cost to obtain resonant frequencies, average response levels and mode shapes. Paper gives details on construction and testing.

CHAPTER 10

WEAR TEST PROGRAM PROGRESS REPORT

A. INTRODUCTION

During the preliminary analysis of the pumps for the Tehachapi Pumping Plant of the California Aqueduct it was determined that insufficient data was available for optimum selection of materials for the metal surfaces subjected to erosion and corrosion by the pumped fluid. The erosion and corrosion that will be encountered in these pumps will be influenced by the high pressure differentials, high surface velocities, and the particular chemical and physical characteristics of the water.

In order to provide useful data for a reliability study to be conducted as a part of the research and development assignment and to develop information for a prototype pump specification, the following program was formulated:

1. Study existing pump installations in the United States and in Europe to determine:

    a. Maintenance experience

    b. Failure Modes

    c. Failure rates

2. Study available literature on wear due to erosion and corrosion.

3. Perform a material wear testing program.

4. Prepare a report presenting findings and experimental results.

5. Incorporate test results and failure and behavior data in the Pump Reliability Study.

6. Prepare prototype pump materials specifications.

B. DISCUSSION

Continuing review of literature and data on wear due to erosion and corrosion is being conducted by DMJM. This review has included study of some of the major hydraulic machinery installations in the United States and Europe. Additional specific data must be obtained, to supplement available data for the particular conditions under which the pumps will be operating at the Tehachapi Pumping Plant.

1. The Problem

The Tehachapi Pumping Plant of the California Aqueduct will represent a major capital and operating cost to the Department of Water Resources. In fact, the pumps in the Tehachapi Crossing must produce combined heads and flow rates that are greater than those in any existing water system. Every effort must be made to assure that the design will provide a high degree of reliability. Due to the high cost of the pumping plant and its components and the high degree of reliability and availability required in the operation of the plant, maintenance requirements and down time must be minimized.

Pump components will wear due to rubbing contact, erosion by the pumped water and cavitation erosion. Corrosion will occur in conjunction with these types of wear. Three major factors influencing wear rates are:

a. Pressure

b. Velocity

c. Water Quality

High pressure discharge across wearing rings and balancing disks and high fluid surface velocities will occur in the Tehachapi pumps simply due to the high lift, large flow rate requirement. Local pressures throughout the pump will depend on the arrangement and design of water passages, seal rings, the number of stages, etc. The corrosive and erosive nature of the water will depend on its PH, hardness, mineral constituents, amount of suspended solids, and the shape and hardness of the suspended solids. These elements of water quality cannot be accurately related to the hardness value of 440 parts per million guaranteed to the user agencies by the Department of Water Resources, and so more detailed knowledge of water quality must be employed in determining wear expectations for the pumping machinery.

In order to produce pumps that will have minimum maintenance requirements, expected wear rates as influenced by performance requirements, mechanical design, and materials selection should be determined. With performance requirements given and mechanical design following time proven experience, then improvements in wear resistance are most likely to be achieved through utilizing detailed knowledge of material and water quality factors. The purpose of the wear test program is to obtain wear data on various materials when exposed to waters similar to the Tehachapi water and under conditions of pressure and velocity that may be expected in the Tehachapi pumps.

2. Availability and Use of Existing Wear Data

A certain amount of wear data is available in the literature and more has been obtained from the operating experience of pumping installations.

a. Pumping Installations

Operational experience, plant operator opinions, manufacturers' opinions, and maintenance and overhaul records have been gathered from European and American pumping and hydro-electric installations for determining wear rates due to corrosion, erosion and cavitation. Unfortunately, this data has not often been precisely recorded or reported by the contributor, and careful review and evaluation were required. As best as possible, the data was analyzed to relate the causes, nature, frequency and extent of failures to: the quality of the water handled; the materials used; the velocities involved; the operating pressures; and the other basic pump parameters. Results of this study are given in the survey reports and reliability reports.

b. Previous Wear Test Data

Previous testing and research programs have produced some wear test data. However, the test conditions have seldom been similar to the Tehachapi situation. For example, the Detroit Edison Co.* ran high pressure tests with good quality, 250°F boiler feedwater that does not compare with aqueduct water. In these tests a proportional increase in the wear rate was observed for some materials when the pressure differential

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* "Corrosion-Erosion of Boiler Feed Pumps and Regulating Valves", pp. 389-403 of ASME Transactions, May 1947.

and flow velocity was increased but no logical relation between wear rate and pressure differential was indicated for others. In view of the low solids content of the water utilized for the Detroit Edison tests it is not possible to apply relative wear rates directly to the Tehachapi pumps. The higher expected suspended solids concentration of the California Aqueduct water will tend to increase wear due to particle erosion.

More recent information concerning wear has been presented in an ASTM publication (Special Technical Publication No. 307) entitled "Symposium on Erosion and Cavitation". Papers in this collection give results of accelerated cavitation tests, wear erosion tests, and sand erosion tests. The results are very interesting but again, due to variation in water quality and operating conditions, conclusions cannot be transferred to the Tehachapi problem. In fact, peculiarities of the results point up the need for specific wear testing simulating Tehachapi conditions.

3. Cavitation Erosion

Wear due to the presence of cavitation normally occurs on impeller vanes near the inlet and may occur at other places within the pump. Sometimes cavitation erosion and wear due to high velocity, silt laden water may occur together and they are not always separately identified. Most materials that are wear resistant to sand erosion are generally resistant to cavitation erosion. There are, however, some differences in wear resistance as reported in the literature. Cavitation erosion rates are known to depend on the water quality. PH value, salt content, etc., affect cavitation erosion. The Tehachapi pumps will probably be subject to some cavitation erosion unless a greater than usual pump submergence is employed. Therefore, any program to study wear should logically include study of cavitation damage.

The plant investigations and literature studies are including the subject of cavitation damage. The experimental program includes accelerated cavitation tests to help identify the kinds and relative amounts of erosion that may occur at Tehachapi and to determine the best materials of construction considering all types of wear.

C. EXPERIMENTAL WEAR TEST PROGRAM

To aid in selecting the optimum materials for the Tehachapi Pumps, DMJM is conducting an experimental wear test program with the assistance of DWR personnel. This program will evaluate and demonstrate the reliability of preferred materials to resist wear due to water erosion,

cavitation erosion and corrosion at the pressure differentials which will exist in the prototype pumps.

A major factor affecting the wearing rate is the quality of the water being pumped. Solid particles in the water traveling at high velocities may tend to penetrate the oxide layer on some metals and increase the wearing rate. It is, therefore, essential to use test water similar to that expected at the Tehachapi Pumping Plant. Conferences with DWR personnel from the Division of Operations led to the opinion that the best correlation could be obtained by conducting the wear test program at the intake to the Tracy Pumping Plant on the Delta-Mendota Canal.

Two types of testers for water erosion are being used in the test program —stationary testers and rotating testers. High pressure water is supplied to the testers by supply pumps taking water from a canal; the pressure differential across the testers is controlled by an inlet pressure regulator and an outlet relief valve.

The test program will provide information on the relative wear characteristics of various materials when subjected to water erosion and corrosion, such as that which will occur in the impeller and casing wearing rings of the prototype pumps. This information will be specifically applicable to wearing rings and in a general way also to the impellers and pump casing.

The stationary type of tester is used to evaluate different classes of materials. Certain selected materials are then tested in the rotative tester to determine wear rates comparable to those that will occur in the Tehachapi pumps.

Accelerated cavitation damage tests utilizing an ultrasonic (magnetostrictive) device will be conducted with water samples taken at Tracy. The information will be directly comparable to the high pressure testing so that behavior of materials with both types of erosion can be determined.

Tests will be repeated for each of four seasons to determine the effects of seasonal variations in water quality. The following materials are being tested initially:

1. Government bronze 88-10-2.

2. 12% chrome stainless steel 450-500 B.H.

3. 18-8 stainless steel (A.I.S.I. 304)

4. 18-8 with 2% molybdenum (A.I.S.I. 316)

5. 17-4 pH stainless steel 500 B.H.

6. Hard-surfaced material, 12% chrome with Stellite overlay.

7. 12% chrome stainless steel with o.007" chrome plating.

8. 12% stainless steel with Epoxy coating.

D. FACILITY AND EQUIPMENT

Drawing A-1 shows the facility for wear testing. It features a high pressure pump for each of the tester types (stationary and rotating) with associated regulating valves and instruments. Each tester type can be operated independently of the other. Details of the test set-ups and testers are shown in Drawings M-1, M-2, M-3 and M-4. (Figures 10-1, 10-2, 10-3, 10-4 and 10-5).

1. Stationary Tester Operation

Eight stationary testers are employed to permit simultaneous testing of eight materials. Water, pumped from the canal at high pressure, is directed to the center of each test specimen pair where it jets in opposite directions from the center along the .020 x .25 slots. Erosion along the slots is measured by carefully cleaning and weighing the specimen periodically during the test.

Tests will be run at a pressure of 216 psi (500 feet of water) for 500 hours and at 432 psi (1,000 ft. of water) for 1,000 hours. Supply pressure, differential pressure, and flow rate are monitored daily and water quality data is taken as needed. Initially, water sampling is being conducted on a rather frequent basis, but it is anticipated that less samples will be required after patterns have been established. Test specimens are cleaned and weighed at approximately 250 hour intervals. Photographs of specimens are taken periodically.

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[1] Figures are at the end of this chapter.

Normally, all specimens will be changed after each 500 hour or 1,000 hour test is completed. Exceptions may be made if a specimen shows excessive wear prior to completion of the test run; in this case, individual testers may be shut down or the specimens may be changed prior to the completion of the test. As testing progresses, the need for program modifications will appear. Other exceptions to the plan may be made to extend the test duration for selected outstanding materials. Special attention will be given to any specimens tested for extended durations to avoid confusing test conditions.

2. Rotating Tester Operation

In the rotating tester, a stationary and a rotating ring constitute a specimen pair. These rings have a clearance diameter of 8 inches and the inner ring, rotating at 3,450 rpm, will be a peripheral speed of about 720 feet per minute (121 fps) which approximates the peripheral speed of the wearing rings of a prototype pump. Each of two testers accommodates two sets of rings such that four kinds of materials can be tested simultaneously. High pressure water is introduced between the two ring sets and passes through the ring clearances to a common discharge pipe. The radial clearance between the rings will be .010 at the beginning of a test. Wear will be determined by weighing the rings. These testers will give wear rates approximating the prototype pumps and the testing will not be of an accelerated type.

Testing procedures are similar to those used for the stationary tests except fewer materials will be used. Results of stationary tests are being used to guide the rotating tests.

3. Cavitation Erosion Testing

Accelerated tests will be conducted utilizing a commercially available ultrasonic transducer and power supply. Specimen tips made from materials to be tested are simply screwed into the end of the transducer. The power is turned on with the tip submerged a fraction of inch in water. Weight changes are the measure of cavitation damage. This type of test only requires about two hours of significant results and therefore, cavitation erosion tests are being conducted as the time becomes available during the high pressure testing.

E. STATUS OF EXPERIMENTAL PROGRAM

The wear test facility has been in operation since 16 December 1964. The 500 hour test at 500 foot head has been completed on the first group of stationary specimens as well as the 1,000 hour test at 1,000 foot head on the second group of stationary specimens. This completes the stationary program for the winter season.

As of 15 March 1965, the rotating specimens have logged approximately 480 hours at the high head differential pressure. The rotating program has been delayed due to late arrival of test specimens, mechanical difficulties in the testers and problems with the supply pump's motor.

Detailed test procedures have been prepared with copies distributed to all personnel concerned with this program. A copy of these procedures can be found in the Appendix at the end of this chapter.

During the period of testing water samples have been taken and sent to the Bryte Laboratory Facility of the Department of Water Resources for analysis. The entire responsibility of selecting the type of analysis per each sample taken has been made by a representative of this laboratory.

F. WEAR TEST RESULTS

The results of the Winter season tests conducted on the stationary specimens are exhibited in Figure 10-6 for the 500 foot differential pressure and Figure 10-7 for the 1000 foot differential pressure. These curves demonstrate comparison wear rates between the materials tested. It is too early in the test program to discuss or draw any conclusions, however, the Stellite coated specimen and the hard chrome-plated specimen are shown to have the least rate of wear.

Figures 10-8, 10-9, 10-10 and 10-11 show the results of water analysis conducted by Bryte Laboratories on water samples taken during the testing period. Figures 10-12, 10-13, 10-14 and 10-15 show actual data taken at the test facility site or analyzed at Bryte Laboratories while water test equipment is not available at the test site.

Typical data sheets displaying the actual test data currently being recorded by the test monitor are shown in Figures 10-16, 10-17 and 10-18.

Figure 10-19 is a form designed to log the many photographs taken of the specimens during their inspection periods. The rotating specimens have not

yet completed their first 1000 hour test, and there are no test results to report at this time.

APPENDIX

WEAR TEST PROCEDURES

INTRODUCTION

These procedures have been formalized in order to insure consistent acquisition of data for the wear test program. Procedures have been divided into four categories. The responsible supervisors and authorized working personnel have been designated for each category. Mr. Dick Burge will act as overall test activities coordinator for DMJM:

	Test Category	Supervisors	Active Working Personnel
1.	Test Monitoring	Jack Wyatt Dick Burge	Steve Gentry (Test Monitor)
2.	Water Analysis	Jim Morris Jack Wyatt	Steve Gentry Bryte Lab Staff
3.	Test Specimen Care	Jack Wyatt Dick Burge	Bryte Lab Personnel Designated by Jack Wyatt
4.	Facility Maintenance	Jack Wyatt Dick Burge	Steve Gentry

If changes in these arrangements should occur, all supervisors should be notified.

GENERAL

All data will be connected primarily through the process of dating. Data acquisition forms will relate to each other by the dates and also by "Season" and "Test Type". There will be four seasons:

1. Winter
2. Spring
3. Fall
4. Summer

and two test types:

1. 500 ft. head (pressure)
2. 1000 ft. head (pressure)

The appropriate season and test type will be marked on each data form as:

Season: Winter   Test Type: 500

All data sheets for a like season and test type will form a set from which an analysis can be made and a report can be prepared. Also, there are two kinds of tests; rotating and stationary. Thus, when the program is complete, there will be eight sets of data for each of the two kinds of tests, or 16 sets of data all told. However, some of the water analysis data will be common to each kind of test.

Every attempt will be made to keep both rotating and stationary tests on like 250 hr. intervals, in order to minimize trips to the facility and the amount of water samples to be taken.

Water samples will be consecutively numbered, as will be photos, for the extent of the program. If a water sample is lost or not analyzed, a note to that effect will be made on the data form. Whether water samples are used only at the field facility for daily tests or are sent for further analysis to Bryte Lab will not effect the numbering. There simply will be many sample numbers that are not sent to Bryte. (See Water Analysis Procedure).

All photos will have a separate number and the photo log will correlate with other data by date and photo description. A set of data will consist of one or more sheets of the following forms:

Stationary Tester Data Sheet	S-1
Rotating Tester Data Sheet	R-1
Field Water Analysis Data Sheet	W-1
Laboratory Water Analysis Data Sheet	W-2
Photo Log Sheet	P-1

and prints of all photographs logged.

Originals or direct copies of all data reporting sheets will be transmitted to DMJM as soon as possible, after the test is completed for incorporation in reliability analysis and report preparation.

SPECIMENS

Specimens (both rotating and stationary) are marked with a dash number and serial number, thus:

-5S3

The dash number -5 identifies the material and the serial No. S3 is the third piece of that particular material. Eight pieces of each material and specimen part (S1 to S8) have been fabricated in the initial specimen order.

With the stationary specimens, parts with like dash number and serial number are used together as a set. Thus, a top and a bottom, both with the Code -1S1, are installed in the same tester.

In the rotating tests, materials may be fixed; one material for the stationary ring and another material for the rotating ring. Therefore, rotating specimen sets may have different dash numbers and may have different serial numbers, for example:

Stationary ring -1S1
Rotating ring -2S3

The material versus dash number code is given in the following table:

Dash No.	Material & Treatment
-1	Bronze (88-10-2) Hyd. Inst. Grade CB-5
-2	Type 420 Stainless Steel (12-14% chrome), 450-500 B.H.
-3	Type 304 Stainless Steel (18-8)
-4	Type 316 Stainless Steel (18-8 - 2Mo)
-5	17-4 PH Stainless Steel 375-425 B.H.
-6	Stellite No. 6 weld overlay on Type 410 Stainless Steel
-7	Hard Chrome Plate on Type 410 Stainless Steel
-8	"SPECOAT EPOXY" coating on Type 410 Stainless Steel

(Materials -1 to -4 were taken from centrifugally cast stock.)

1. Test Monitoring Procedure

A. General

Following the initial installation of test specimens, or subsequent installation after an inspection period, the tester data and certain water data will be taken. The data will be read daily (except when not practical on Saturdays, Sundays, and Holidays) and will be recorded on Data Sheets S-1, R-1 and W-1. Although tester data is not expected to change much on a daily basis, recording it will give a continuous record and the daily visit will constitute an operational inspection, and opportunity to adjust pressure regulators, etc. Any unusual findings should be immediately reported to one of the test supervisors and recorded on the back of a test data sheet (R-1 or S-1). Use additional sheets if comments become lengthy.

B. Data Period

The testers will operate for approximately 250 hr. intervals between inspections, or approximately 10 days. (Where weekends and holidays interfere, inspections will take place on the nearest work day to 250 hours.) The inspection data is entered on the tester data forms, S-1 and R-1, and so a new sheet for S-1 and R-1 (and also W-1) will be started after each inspection.

C. Data Recording and Equipment Adjustments

(1) Testers

Data is to be entered on the forms S-1 and R-1. A sample set is attached and is reasonably self-explanatory.* Daily readings will be filled out completely. On S-1 and R-1, Temperature, Flow Rates and Pressures can be read from the gauges quite rapidly, and the Differential Pressure calculated by; (Inlet Header Pressure - Outlet Header Pressure). The

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* Data sheets have been eliminated in lieu of actual data on forms exhibited in Section F.

Differential should be:

Test Type	Diff. Pressure
500	216 psi $\pm$ 5 psi
1000	432 psi $\pm$ 10 psi

As the specimens wear, the flow rate will go up and the Differential Pressure will tend to go down. In order to prevent the Differential Pressure from dropping, the pressure regulators on each tester system should be adjusted to return the Differential Pressure to the setting specified above. This adjustment should be made each day if necessary, if the Differential Pressure is out of the tolerance shown above. If the Differential Pressure cannot be brought into the tolerance zone, leave the tester system in operation but notify a supervisor. If an adjustment is made, the pressure gauge readings prior to the adjustment should be recorded and the pressure gauge readings, after the adjustment, should be recorded under a slashed line in the same data space on the form -- see samples attached. Similarly, the flow rates should be repeated.

In the event a flow rate reaches the maximum reading on a meter, the supervisor should be notified. A special inspection may be made by the supervisor or the individual tester may be shut off. If a tester is shut off, an adjustment in the Differential Pressure will probably be required.

In changing from a 500 ft. head test to a 1000 ft. head test, or vice versa, a spring must be changed in the Baily regulator in the rotating tester system in order to put the regulator control in the correct range. Special instructions for the spring change will be given to persons involved.

(2) Water Sampling

A water sample, suitable for the daily water analysis, should be drawn after reading the gauges. Data is to be entered on Form W-1. Daily water samples should be drawn from a bleed valve on the header of either the stationary or rotating tester system. The valve should be opened and allowed to run freely for 30 seconds to a minute, before the sample is taken in order to clear trapped sediment from the valve and line. See Water Analysis Procedure for further sampling instructions and analytical requirements.

2. Water Analysis Procedure

A. General

Water analyses will be conducted in two parts: (1) Field Analysis and (2) Laboratory Analysis. Field Analysis will be maded as part of the Test Monitoring and will follow the schedule set forth in the Test Monitoring Procedure. Field data will be recorded on the FIELD WATER ANALYSIS Data Sheet, W-1.

Laboratory water analysis will be conducted at the Bryte Engineering Laboratory, DWR, and will be performed on samples every 250 hours (approximately 10 days). More or less frequent lab analysis may be made according to the directions of the supervisors. Results will be recorded on Laboratory Water Analysis Data Sheet W-2.

B. Sample Numbering and Handling

Samples will be numbered consecutively and dated. The dates will relate the results to other phases of the testing. As several samples could be taken in a day, the numbering of samples is required. In certain tests, water may be drawn and examined without an actual storage period in a bottle; however, a sample number will still be assigned, corresponding to the data entered on the data sheet. In fact, several separate water "parcels" may be examined for different items and use the same sample number. When a water sample is bottled for storage or delivery to Bryte, it will be labeled with sample number and date. Several bottles drawn at the same time may be labeled with a number and letter, as 17-A, 17-B, etc. If a sample is taken from a different source (river as opposed to tester header) or at a significantly different time of day, it will receive a new number.

All sample numbers will appear on the Field Data Sheet, W-1, as all samples will be subjected to the Field analysis. Samples sent for the more extensive Laboratory Analysis will be noted on W-1 by the "Date Sent to Lab" column. Lab sample analytical results will be reported on the Laboratory Water Analysis Data Sheet, W-2. Original work sheets used in arriving at final results should be attached to the W-1 and W-2 sheets.

C. Analytical Procedures * (outline only given)

(1) Sample Collection

(a) Cleaning sample container - types of containers.
(b) Collection methods (according to intended analytical use).
(c) Recording, labeling, storage and handling (time limits).
(d) Collection Schedules and Assignments.

* Reference: "Standard Methods of Examination of Water, Sewage and Industrial Wastes". APHA.

(2) Field Analysis (Sample taken and work performed by "Field Monitor").

(a) Equipment (by type of test)
(b) Detailed test procedures (by type of test)

I Turbidity (daily)
II Ph (daily)
III Dissolved oxygen (daily)
IV Carbon dioxide (daily)
V Specific conductance (daily)

(3) Laboratory Analysis (Samples taken by "Field Monitor" or "Specimen Inspector"; work by Lab. Staff)

(a) Equipment (by type of test)
(b) Detailed test procedure (by type of test)

I Total dissolved solids (250 hr.)
II Suspended Solids (250 hr.)
III Suspended particle analysis

a. Particle size
b. Particle shape
c. Particle hardness

Indication or Average (250 hr.)

IV	Calcium	(250 hr.)
V	Sodium	(250 hr.)
VI	Silica	(250 hr.)
VII	Potassium	(250 hr.)
VIII	Magnesium	(250 hr.)
IX	Alkalinity	(250 hr.)
X	Chlorides	(250 hr.)
XI	Sulphates	(250 hr.)

3. <u>Procedure for Handling Test Specimens</u>

A. <u>Initial Clearance Measurement</u> (Rotating Specimens Only)

Prior to installation in the rotating testers, the radial clearance of each "running set" of specimens shall be measured and recorded. The method employed will be as follows:

(1) Pair off running set of specimens and place on a flat surface.

(2) Squeeze and hold set together in one spot and measure with wire gauges and the maximum clearance produce in the diametrically opposite site.

(3) Divide this measurement by two and record as the installed radial clearance in space provided on Data Sheet R-1. The result will be approximately $\frac{.020''}{2} = .010''$.

B. <u>Initial Cleaning</u>

Each specimen shall be cleaned thoroughly by first immerging in Benzol and then Acetone. Scrubbing with a non-metallic brush and agitating the solutions will expedite this operation. Specimens shall be handled with laboratory tongs or gloves to prevent solution contact with hands.

C. <u>Drying</u>

Surplus solvent shall be removed by shaking and/or blotting with paper towels. The specimens shall be placed in a desiccator for a

period of 30 minutes minimum. Laboratory tongs shall be used to handle the specimens, taking care so as not to scratch or otherwise damage them.

D. Inspection

Because of the requirement of precision weighing, each specimen shall be microscopically inspected to prove the effectivity of the cleaning process. If necessary, the cleaning process shall be repeated until all foreign material is removed. Other solvents may be used only if the following conditions are satisfied:

(1) Establish proof that the chemical or solvent selected will not remove or change properties of the specimen's material.

(2) Re-wash the specimen in Benzol and Acetone after employing the selected technique.

(3) Write a full description of the exact process and chemicals used, along with the specimen serial number on the reverse side of the data sheet, S-1 or R-1.

E. Weighing

After proper cleaning and drying, each specimen shall be carefully weighed, using a laboratory analytical balance. The stationary tester specimens shall be weighed to an accuracy of plus/minus one-tenth of a milligram ($\pm$ .0001 grams). The rotating tester specimens shall be weighed to an accuracy of plus/minus 10 milligrams ($\pm$.010 grams). Standard laboratory practices and written instructions by the balance manufacturer shall be employed during this operation, including periodic calibration checks of the balance. The weights of each specimen shall be recorded in the space provided on the test data sheet, S-1 or R-1, coordinated with the specimen serial number.

F. Photographing

The wearing surfaces of each specimen shall be photographed, using a camera attached to a microscope. A log of each photo taken will be kept on Data Sheet P-1. Whenever practical, a slip of paper with the serial number and date written on it will be placed in the field of the photograph. When roll film is used, the roll film number will be recorded. As these numbers do not always match the camera operation, roll film will be

processed uncut in order to retain the sequence of photos and the processor shall number prints in sequence, corresponding to the film. Correlation with the photographic log sheet P-1 can then be made.

Initially, there will be one close-up photograph of the specimen's wearing surfaces with a magnification of 1:1. Additional photographs, with higher magnification showing other specific details of the wearing surfaces, may be made as required. After the initial photographs have been viewed and found to be satisfactory the magnification ratio shall be fixed for subsequent photographs of like specimens for the purpose of consistent comparison. In the case of the rotating tester specimens, due to their size, a close-up photograph of only a section of the wearing surface shall be taken. After a period of testing, the section chosen shall be in the area showing the most severe wear. Additional photographs of other sections may be taken, if they serve to further clarify or define possible conclusions. Specimens shall be handled with laboratory tongs during the photographing, in order to avoid contamination.

G. Installation in Testers

(1) Stationary Specimens

Specimens are installed following the assembly drawing - DMJM M-2. The specimens are placed in position and the tester assembly completed except for the pressure screw 3 and cap 7 . Nuts on studs should be uniformly tightened to ensure leak tight gasket joints. Pressure screw 3 should be carefully screwed in place and made "snug". It should not be over-tightened to the point of marring the top surface of the specimens. Cap 7 is then positioned and tightened against the top gasket.

When all specimens are installed, the tester water supply is started, a check made for leaks, and then the first set of data is taken. Regulator adjustments are made to obtain the proper Pressure Differential.

(2) Rotating Specimens

Specimens are installed two sets to a tester, as shown in DMJM Drawing M-4. Care should be taken to ensure that the axial position of the specimens is correct. The case, section 1 should be positioned so as to obtain the most nearly uniform radial clearance. Clearance of the front set only should be checked at four positions around the specimens utilizing the "music" wire gauges.

Tester leakage should be checked and the pressure regulator adjusted for the proper Pressure Differential and the first data set taken. If a change from a 500 ft. head test to a 1000 ft. head test is being made, the regulator spring in the Baily regulator will have to be changed. Special instructions will be given to personnel.

H. Re-inspection, Recleaning, Redrying, Reweighing and Re-photographing

(1) Order of Work

At the conclusion of each test phase, each specimen shall be handled according to the following sequence:

(a) Preliminary inspection
(b) Preliminary cleaning and drying
(c) Detailed inspection
(d) Photographing
(e) Cleaning
(f) Drying
(g) Weighing
(h) Reinstallation in tester

(2) Preliminary Inspection (a) above

The objective of this examination is to look for anything unusual that has happened to the specimens during the test period; such as clogged wear passages, broken or loose parts in the testers or rubbing in the case of the rotating tester specimens. A full description of anything seemingly unusual shall be reported on the back of the test data sheet, S-1 or R-1.

(3) Preliminary Cleaning and Drying (b) above

The specimens shall be cleaned and dried essentially, as described in paragraphs B, C and D; however, the objective here is to remove the accumulation of deposits as necessary only to expose paths of wear to be examined in detail and photographed. The specimens shall be further cleaned with precision after the inspection and photographing periods and before weighing.

(4) Detailed Inspection-Photographing (c) & (d) above

Each specimen shall be carefully inspected in detail for wear paths and patterns and the selection of sections demonstrating the most severe damage as subject for photographing. Photograph the interesting areas following the discussion of Section E. At the conclusion of the photographing, the radial clearance of the rotating specimens shall be re-measured and recorded. Refer to Section 3A of this document for the method previously employed. Enter this measurement in two places on different R-1 data sheets: (1) in space provided opposite "removed clearance" on old data sheet; and, (2) in space provided opposite "installed clearance" on the new data sheet to be used in the next 250 hour test period. As the wear surface may now be rough, precision in this measurement should not be expected.

(5) Cleaning, Drying and Weighing (e), (f), (g) above

The cleaning and weighing shall follow the procedures given in Section B, C, and D. However, if scale and corrosion are present, it will probably be necessary to use cleaning solutions such as hydrochloric acid inhibited with antimony chloride. Other solvents may be used subject to the conditions stated in Section C.

(6) Installation in Testers

Specimens shall be placed in the same testers as originally installed. Rotating specimens shall be mounted in the same end of the tester as originally installed.

I. Final Inspection

At the conclusion of a test, the inspection shall be conducted as outlined in Section G, except there will be no re-installation of specimens.

Specimens that have completed tests will be stored in a safe manner, awaiting shipment to DMJM. Used specimens will be held at DMJM for reference in preparing reports.

4. Facility Maintenance

A. General

As experience is gained in the operation of the test facility, more items of maintenance will be recognized and should be added to this list.

B. Inspections and Maintenance

(1) Daily (Test Monitor)

Visual inspection of supply pumps, piping and testers for excess seal leakage, over-heating of seals, bearings, etc. Inform supervisors of any unusual circumstances and note on the back of data sheets.

(2) During Specimen Inspections (250 hrs. - 10 days) (Specimen Inspector)

(a) Inspect disassembled tester parts for -- check gaskets and replace if necessary (both stationary and rotating).

(b) Check supply pump seal for leakage, replace packing if necessary.

(c) Check supply pump motors and pumps for lubricating oil -- add if low.

(d) Clean regulator orifices and filters.

(e) Clean BJ pump seal water filter.

(f) Check cleaning solutions and water analysis solutions. Replace all exhausted solutions. Check dessicants.

(g) Check balance accuracy.

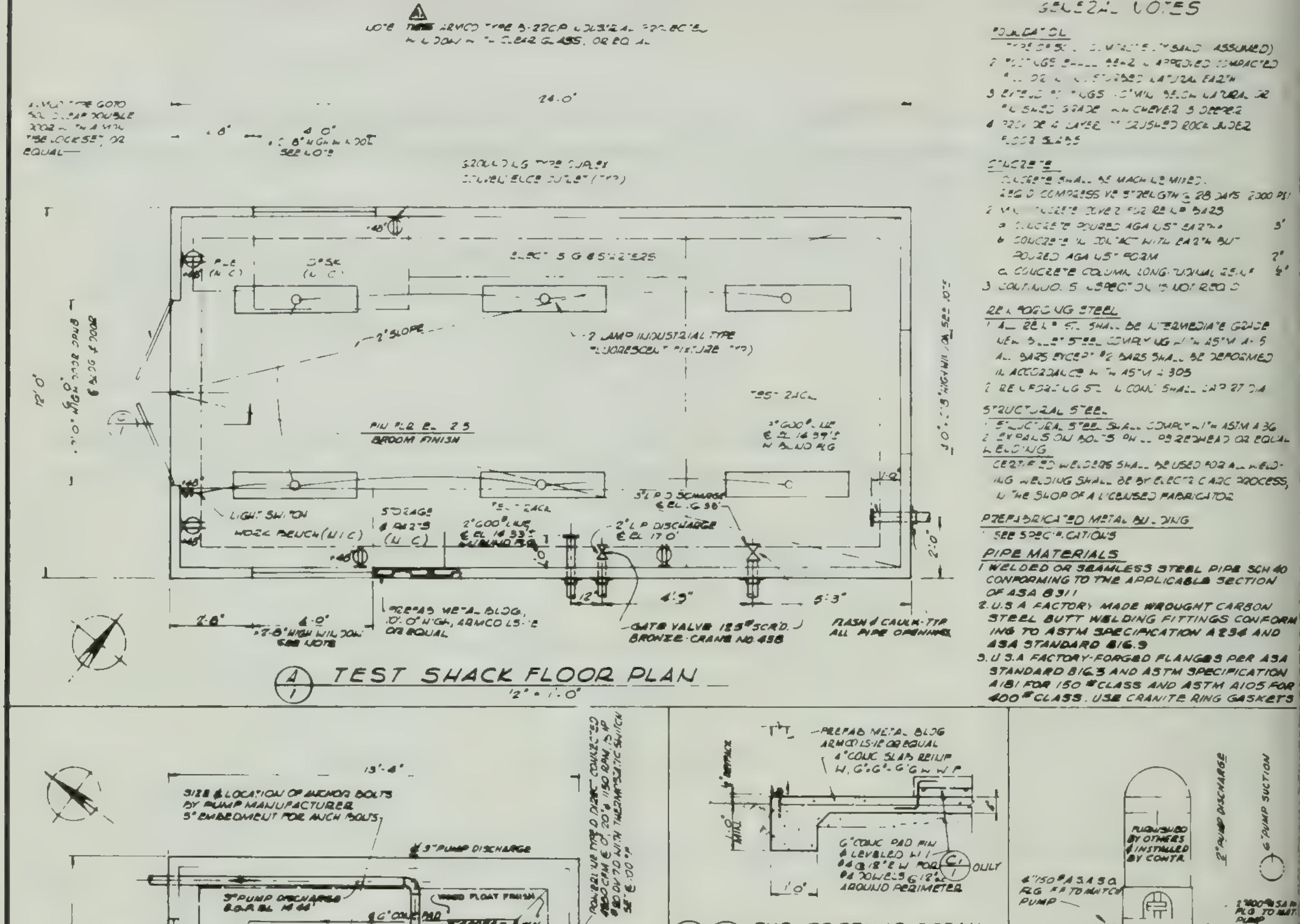
(3) At the completion of a Season

(a) Check stocks of towels and rags, personnel supplies, gasket material, cleaning solutions, dessicant, water bottles, water analysis, chemicals, etc., and replenish.

(b) Inspect all piping and supply pumps and motors, and rotating tester motors for loose mounting, loose parts, evidence of wear, etc.

(c) Inspect pump seal sleeves by removing packing and replace packing as necessary. If sleeves are badly worn, order spares and schedule repair work for an inspection period.

(d) Check pressure gauges, thermometers, pressure switches, valves, flow instruments, etc. Clean and repair or replace as required.



DMJM

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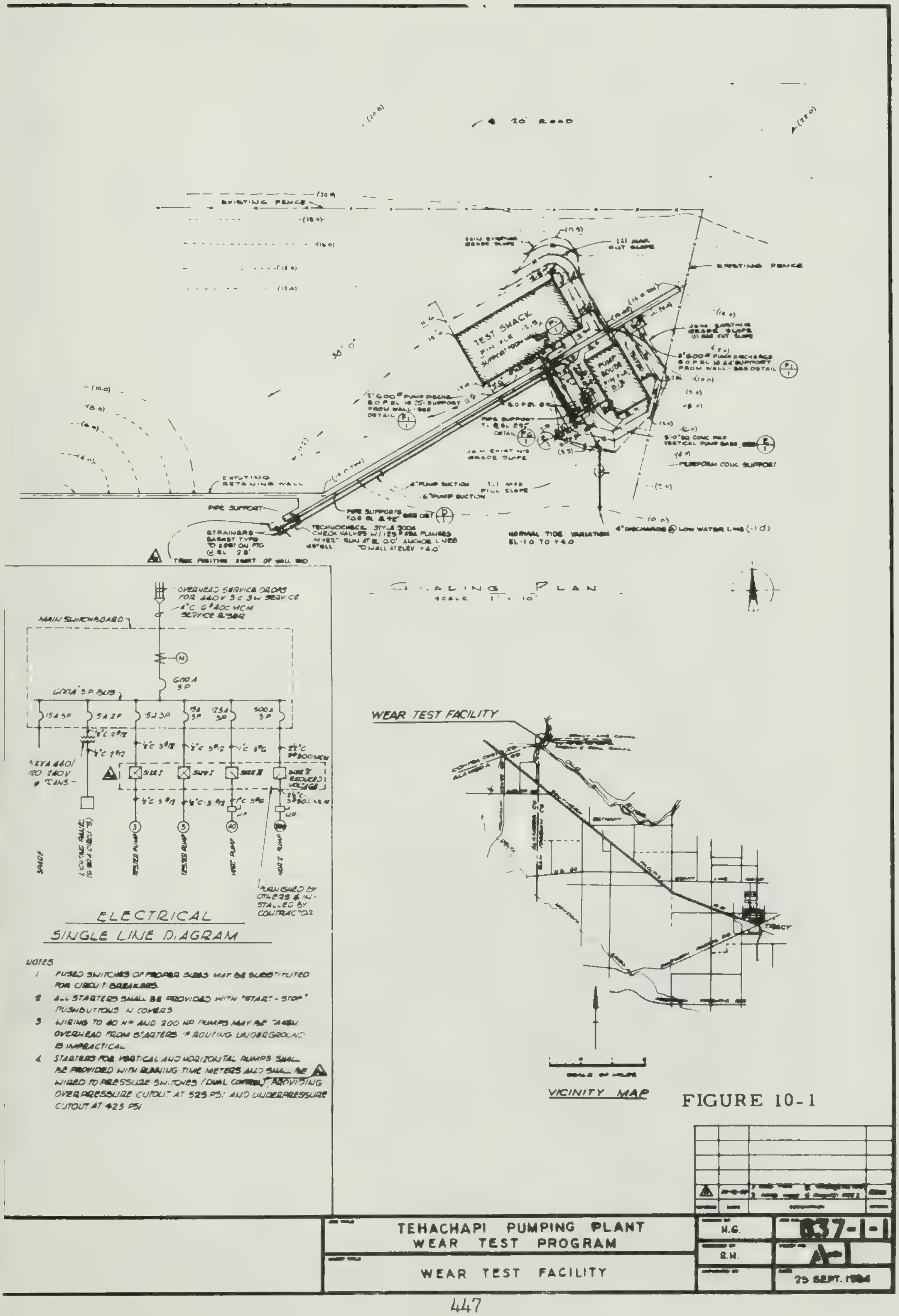


20' ROAD
EXISTING FENCE
TEST SHACK
PUMP HOUSE
EXISTING RETAINING WALL
PIPE SUPPORT
4" PUMP SUCTION
6" PUMP SUCTION
NORMAL TIDE VARIATION EL -1.0 TO +4.0
LOW WATER LINE (-1.0)
GRADING PLAN
MAIN SWITCHBOARD
600A 3P BUS
ELECTRICAL
SINGLE LINE DIAGRAM
NOTES
1 FUSED SWITCHES OF PROPER SIZES MAY BE SUBSTITUTED FOR CIRCUIT BREAKERS.
2 ALL STARTERS SHALL BE PROVIDED WITH "START - STOP" PUSHBUTTONS IN COVERS
3 WIRING TO 40 HP AND 200 HP PUMPS MAY BE TAKEN OVERHEAD FROM STARTERS IF ROUTING UNDERGROUND IS IMPRACTICAL
4 STARTERS FOR VERTICAL AND HORIZONTAL PUMPS SHALL BE PROVIDED WITH RUNNING TIME METERS AND SHALL BE WIRED TO PRESSURE SWITCHES (DUAL CONTROL) PROVIDING OVERPRESSURE CUTOUT AT 525 PSI AND UNDERPRESSURE CUTOUT AT 425 PSI
WEAR TEST FACILITY
TRACY
VICINITY MAP
TEHACHAPI PUMPING PLANT
WEAR TEST PROGRAM
WEAR TEST FACILITY
H.G.
R.H.
A-1
25 SEPT. 1984

FIGURE 10-1

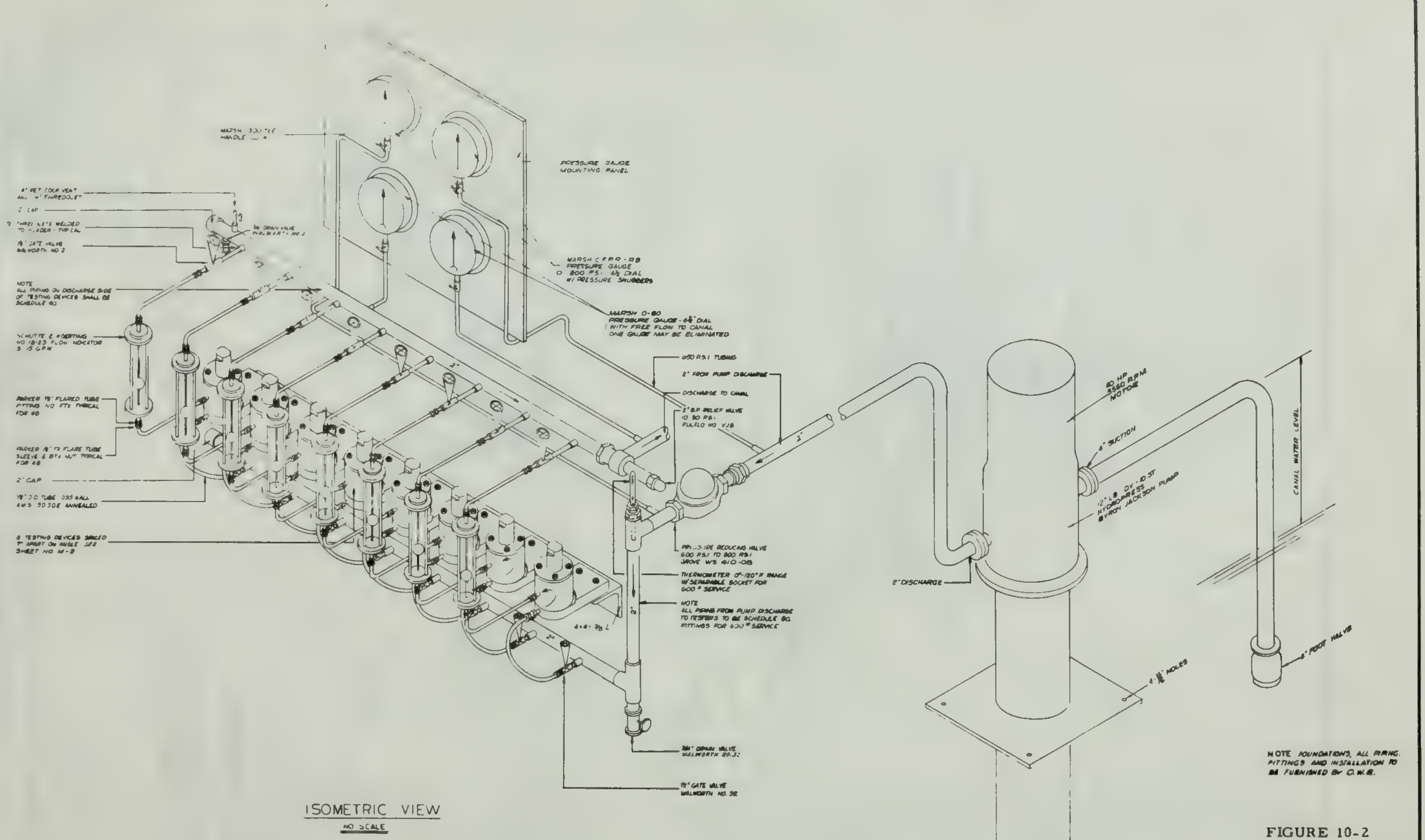
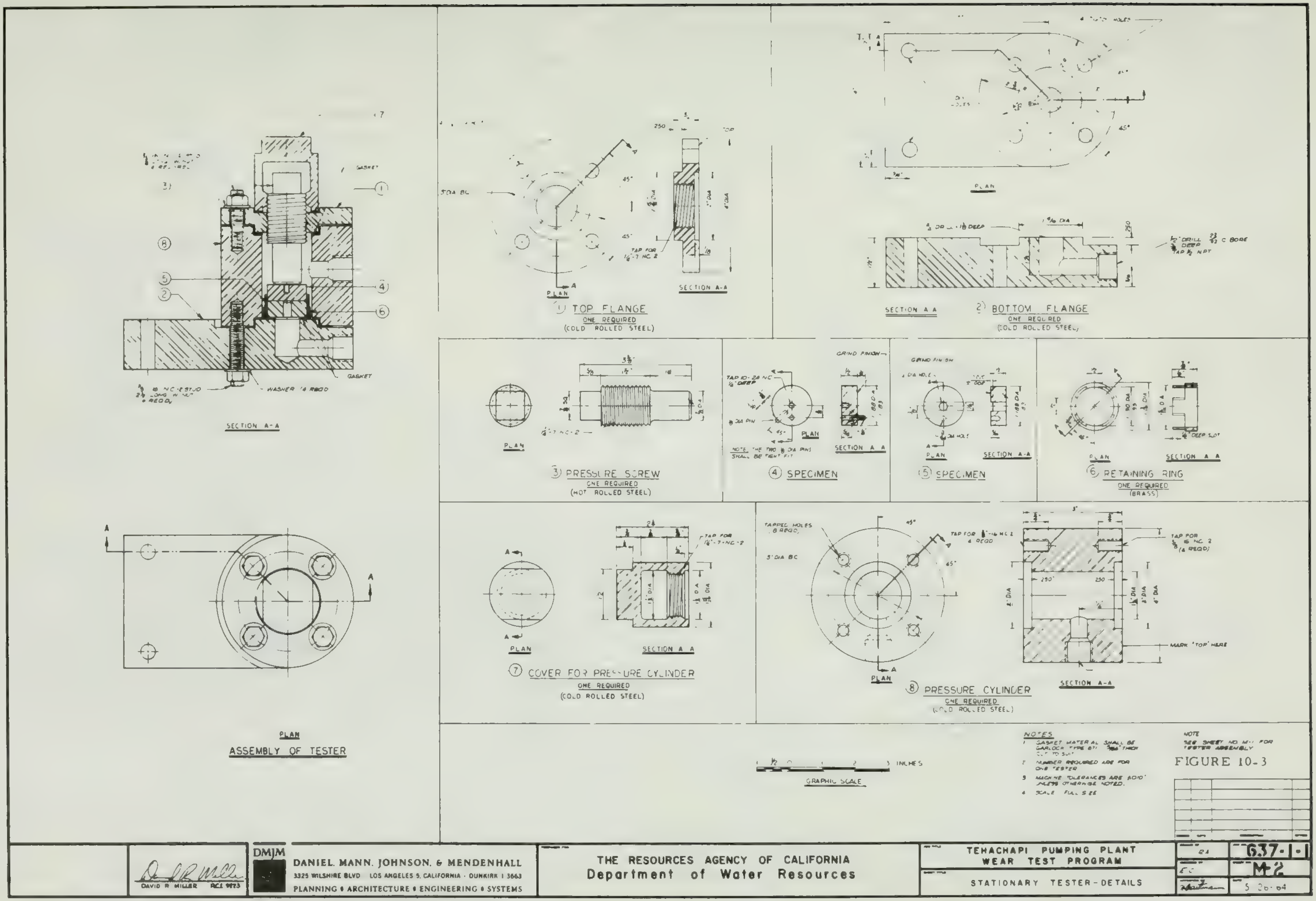


FIGURE 10-2

DAVID R. MILLER RCE 9223	DMJM DANIEL, MANN, JOHNSON, & MENDENHALL 3325 WILSHIRE BLVD LOS ANGELES 5, CALIFORNIA DUNKIRK 1 3663 PLANNING • ARCHITECTURE • ENGINEERING • SYSTEMS	THE RESOURCES AGENCY OF CALIFORNIA Department of Water Resources	TEHACHAPI PUMPING PLANT WEAR TEST PROGRAM STATIONARY TESTER - PIPING DIAGRAM
			637-1-1 M-1 5-26-64


SECTION A-A
PLAN
ASSEMBLY OF TESTER
GASKET
WASHER
1 TOP FLANGE
ONE REQUIRED
(COLD ROLLED STEEL)
2 BOTTOM FLANGE
ONE REQUIRED
(COLD ROLLED STEEL)
3 PRESSURE SCREW
ONE REQUIRED
(HOT ROLLED STEEL)
4 SPECIMEN
5 SPECIMEN
6 RETAINING RING
ONE REQUIRED
(BRASS)
7 COVER FOR PRESSURE CYLINDER
ONE REQUIRED
(COLD ROLLED STEEL)
8 PRESSURE CYLINDER
ONE REQUIRED
(COLD ROLLED STEEL)
GRIND FINISH
MARK 'TOP' HERE
GRAPHIC SCALE
INCHES
NOTES
DANIEL, MANN, JOHNSON, & MENDENHALL
3325 WILSHIRE BLVD LOS ANGELES 5, CALIFORNIA · DUNKIRK 1 3663
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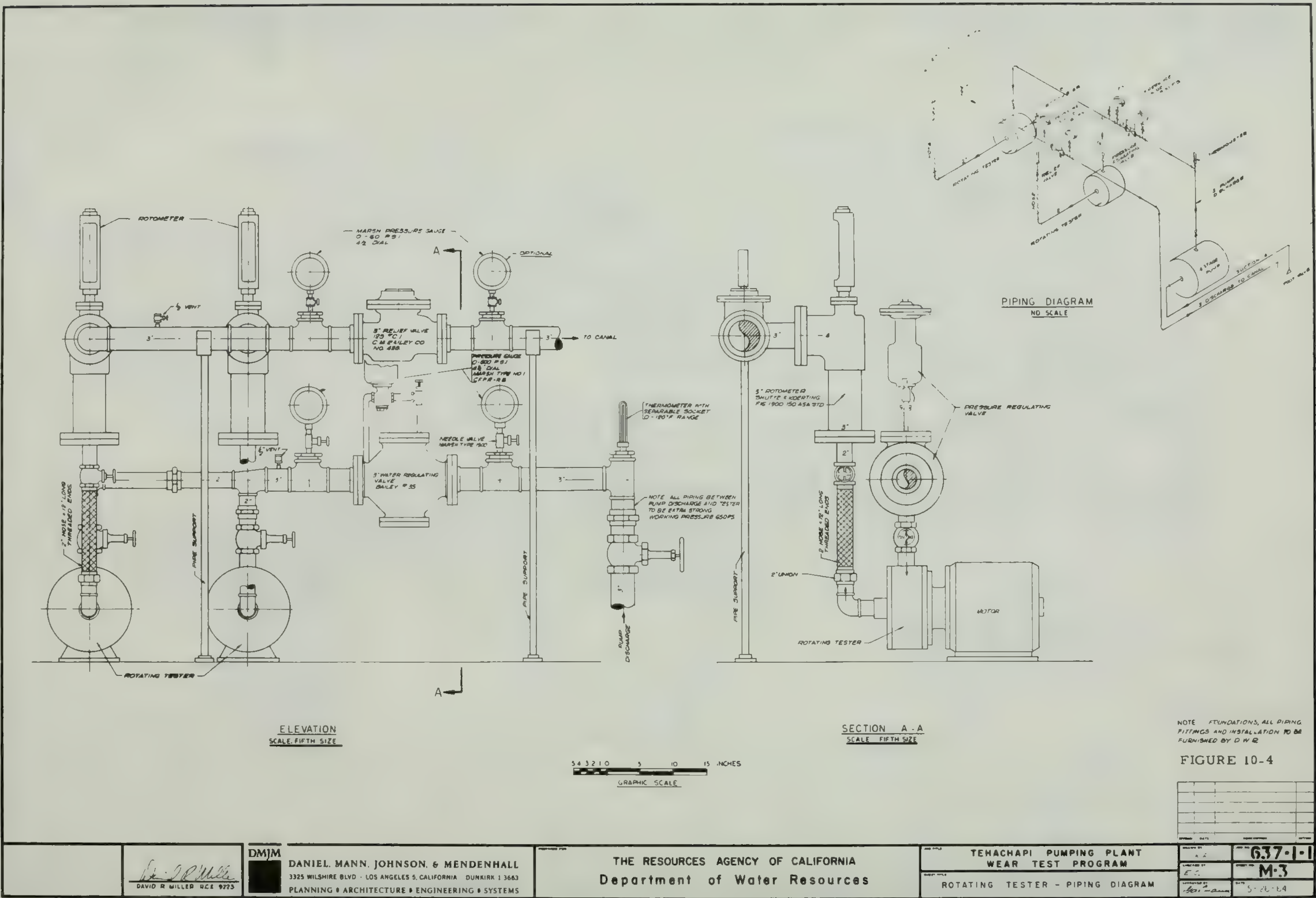
Department of Water Resources
TEHACHAPI PUMPING PLANT
WEAR TEST PROGRAM
STATIONARY TESTER-DETAILS
637-1-1
M-2
DAVID R. MILLER

FIGURE 10-3


ROTOMETER
1/2" VENT
MARSH PRESSURE GAUGE 0-60 PSI 4 1/2" DIAL
OPTIONAL
3" RELIEF VALVE 125 # C.I. C.M. BAILEY CO. NO. 488
PRESSURE GAUGE 0-800 PSI 4 1/2" DIAL MARSH TYPE NO.1 CFPR-RB
TO CANAL
NEEDLE VALVE MARSH TYPE 900
3" WATER REGULATING VALVE BAILEY # 35
THERMOMETER WITH SEPARABLE SOCKET 0-180°F RANGE
NOTE ALL PIPING BETWEEN PUMP DISCHARGE AND TESTER TO BE EXTRA STRONG WORKING PRESSURE 650PS
2" HOSE x 12" LONG THREADED ENDS
PIPE SUPPORT
PUMP DISCHARGE
ROTATING TESTER
3" ROTOMETER SHUTTE & KOERTING FIG 1900 150 ASA STD
PRESSURE REGULATING VALVE
2" UNION
MOTOR
PIPING DIAGRAM
NO SCALE
ELEVATION
SCALE: FIFTH SIZE
SECTION A-A
SCALE: FIFTH SIZE
GRAPHIC SCALE
INCHES
NOTE: FOUNDATIONS, ALL PIPING FITTINGS AND INSTALLATION TO BE FURNISHED BY D.W.R.
DAVID R. MILLER RCE 9723
DMJM
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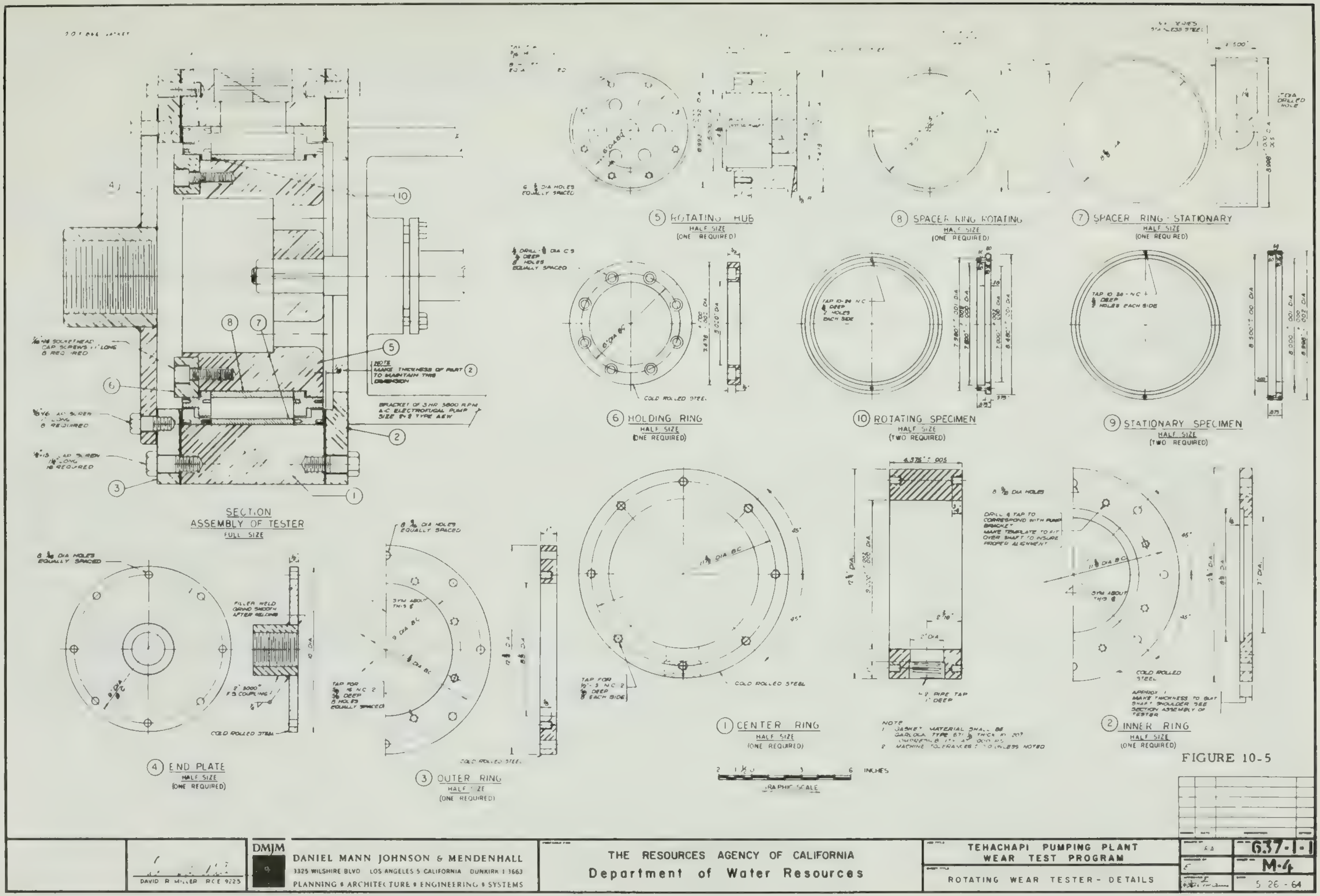
Department of Water Resources
TEHACHAPI PUMPING PLANT WEAR TEST PROGRAM
ROTATING TESTER - PIPING DIAGRAM
637-1-1
M-3

FIGURE 10-4


SECTION
ASSEMBLY OF TESTER
FULL SIZE
(5) ROTATING HUB
HALF SIZE
(ONE REQUIRED)
(8) SPACER RING ROTATING
HALF SIZE
(ONE REQUIRED)
(7) SPACER RING - STATIONARY
HALF SIZE
(ONE REQUIRED)
(6) HOLDING RING
HALF SIZE
(ONE REQUIRED)
(10) ROTATING SPECIMEN
HALF SIZE
(TWO REQUIRED)
(9) STATIONARY SPECIMEN
HALF SIZE
(TWO REQUIRED)
(4) END PLATE
HALF SIZE
(ONE REQUIRED)
(3) OUTER RING
HALF SIZE
(ONE REQUIRED)
(1) CENTER RING
HALF SIZE
(ONE REQUIRED)
(2) INNER RING
HALF SIZE
(ONE REQUIRED)
COLD ROLLED STEEL
GRAPHIC SCALE
INCHES
DAVID R MILLER RCE 9225
DMJM
DANIEL MANN JOHNSON & MENDENHALL
3325 WILSHIRE BLVD LOS ANGELES 5 CALIFORNIA DUNKIRK 1 1663
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Department of Water Resources
TEHACHAPI PUMPING PLANT
WEAR TEST PROGRAM
ROTATING WEAR TESTER - DETAILS
637-1-1
M-4
5 26 - 64

FIGURE 10-5

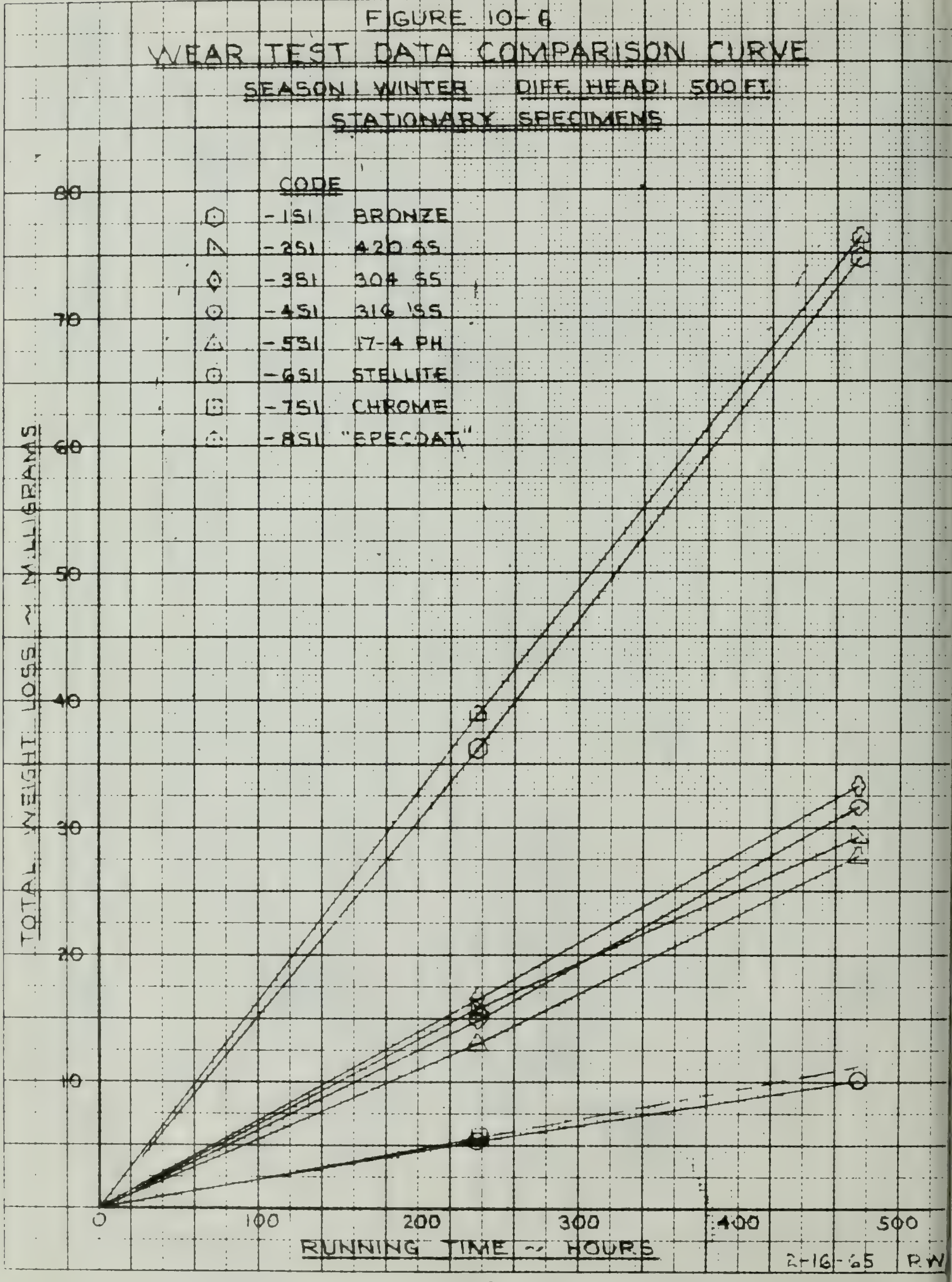
FIGURE 10-6
WEAR TEST DATA COMPARISON CURVE
SEASON: WINTER DIFF. HEAD: 500 FT.
STATIONARY SPECIMENS
CODE
-1SI BRONZE
-2SI 420 SS
-3SI 304 SS
-4SI 316 SS
-5SI 17-4 PH
-6SI STELLITE
-7SI CHROME
-8SI "EPECOAT"
TOTAL WEIGHT LOSS ~ MILLIGRAMS
80
70
60
50
40
30
20
10
0
100
200
300
400
500
RUNNING TIME ~ HOURS
2-16-65 R.W.

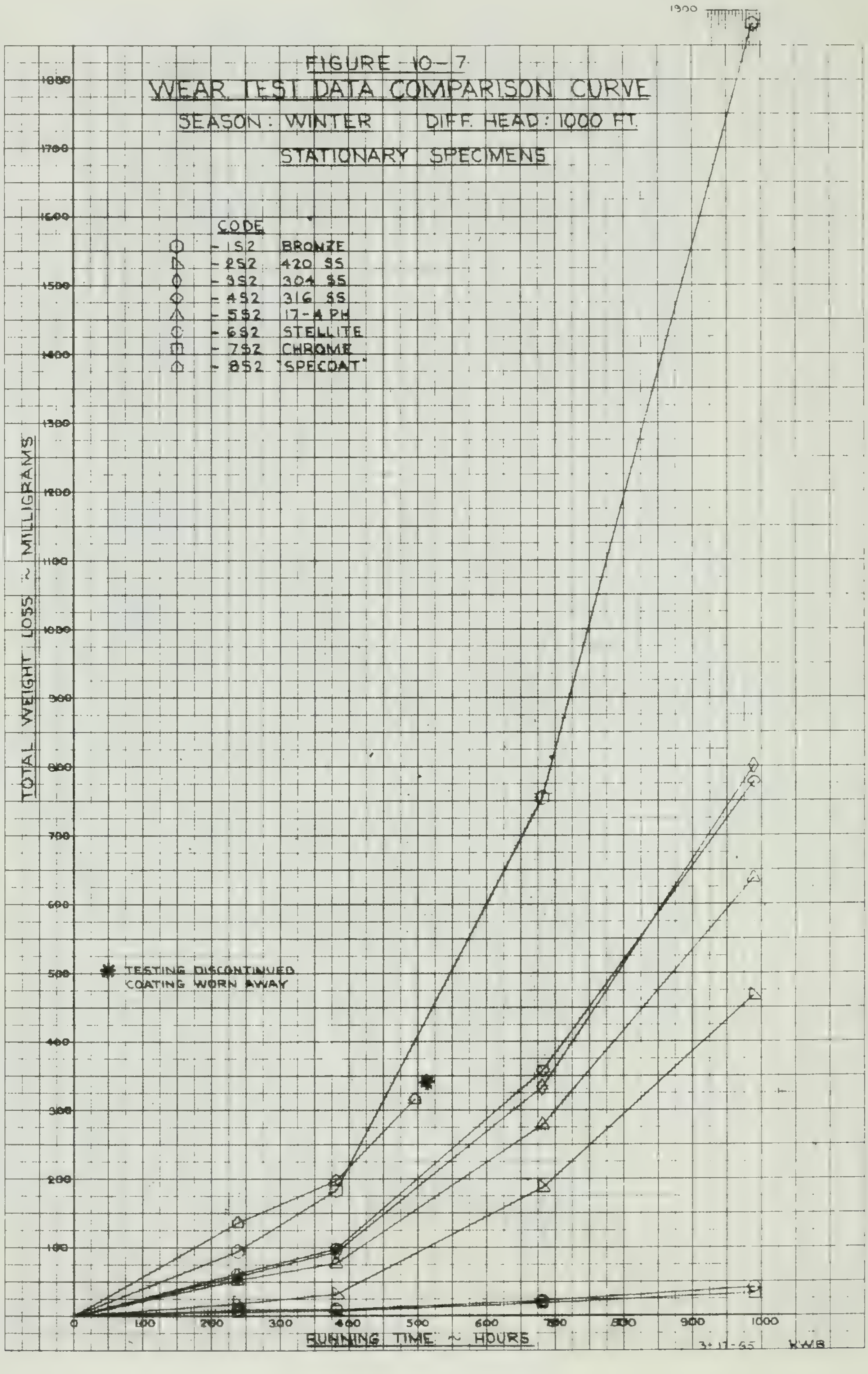
FIGURE 10-7
WEAR TEST DATA COMPARISON CURVE
SEASON: WINTER DIFF. HEAD: 1000 FT.
STATIONARY SPECIMENS
CODE
- 1S2 BRONZE
- 2S2 420 SS
- 3S2 304 SS
- 4S2 316 SS
- 5S2 17-4 PH
- 6S2 STELLITE
- 7S2 CHROME
- 8S2 "SPECOAT"
* TESTING DISCONTINUED COATING WORN AWAY
TOTAL WEIGHT LOSS ~ MILLIGRAMS
RUNNING TIME ~ HOURS
0
100
200
300
400
500
600
700
800
900
1000
1100
1200
1300
1400
1500
1600
1700
1800
1900
3-17-65
KWB

FIGURE 10-8

Sheet 1 of 2

LABORATORY WATER ANALYSIS DATA SHEET

Season WINTER Test Type 500

Sample No.	Date Analyzed	By (Initial)	Total Dissolved Solids PPM	Suspended Solids PPM	*Particulate Data: Size (Range)	Hardness (Ave. & Range)	Shape (General)	Calcium	Sodium	Silica	Potassium	Magnesium	Alkalinity	Chlorides PPM	Sulfates
1	12-24-64	JR												110	
2	12-24-64	JR												142	
3	12-24-64	JR												122	
4	12-24-64	JR												111	
5	12-24-64	JR												115	
6	1-12-65	TM		26										113	
7	1-12-65	TM		34										116	
8	1-12-65	TM		23										115	
9	1-12-65	TM	446	41				35	85		4.1	19	116	120	82
10	1-12-65	TM		64										122	
11	1-12-65	TM		109										38	
12	1-12-65	TM		42										36	
13	1-12-65	TM		79										119	
14	1-12-65	TM		239										21	
15	1-12-65	TM	120	117				9.6	11		2.4	3.6	38	15	10
16	1-21-65	RV		103										50	
17	1-21-65	RV		103										33	
18	1-21-65	RV		87										45	
19	1-21-65	RV		104										37	
20	1-21-65	RV		92										26	

*Supplement with occasional Micro Photograph

Comments:

FIGURE 10-9

LABORATORY WATER ANALYSIS DATA SHEET

Sheet 2 of 2

Season WINTER Test Type 500

Sample No.	Date Analyzed	By (Initial)	Total Dissolved Solids PPM	Suspended Solids PPM	*Particulate Data			Calcium	Sodium	Silica	Potassium	Magnesium	Alkalinity	Chlorides	Sulfates
					Size (Range)	Hardness (Ave. & Range)	Shape (General)	PPM	PPM	PPM	PPM	PPM	as HCO_3	PPM	PPM
21	1-21-65	RV		34										59	
22	2-1-65	SL	133	139				12	17	14	2.3	5.4	47	24	19
23	1-21-65	RV		146										43	
24	2-1-65	SL	240	78				13	42	16	2.7	9.1	60	58	42
25	1-21-65	RV		75										59	

*Supplement with occasional Micro Photograph

Comments:

FIGURE 10-10

LABORATORY WATER ANALYSIS DATA SHEET

Sheet 1 of ___

Season WINTER Test Type 1000

Sample No.	Date Analyzed	By (Initial)	Total Dissolved Solids PPM	Suspended Solids PPM	*Particulate Data Size (Range)	Hardness (Ave. & Range)	Shape (General)	Calcium	Sodium	Silica	Potassium	Magnesium	Alkalinity	Chlorides	Sulfates
26	2-19-65	RV		68											
27	2-13-65	RV		76											
28	2-19-65	RV		92											
29	2-19-65	RV		76											
30	2-19-65	RV		88											
31	2-19-65	RV		87											
32	2-17-65	RV		77											
33	2-19-65	RV		68											
34	2-19-65	RV		74											
35	2-19-65	RV		51											
36	2-19-65	RV		40											
37	2-19-65	RV		50											
38	2-19-65	RV		46											
39	2-19-65	RV		42											
40	2-19-65	RV		43											
41	2-19-65	RV		50											
42	2-19-65	RV		49											
43	2-23-65	RV		69											
44	2-23-65	RV		73											
45	2-23-65	RV		52											

*Supplement with occasional Micro Photograph

Comments:

DA [illegible]

Los Angeles, California

California Pump
Wear Test Program

FIGURE 10-11

Sheet 2 of ___

LABORATORY WATER ANALYSIS DATA SHEET

Season WINTER Test Type 1000

Sample No.	Date Analyzed	By (Initial)	Total Dissolved Solids PPM	Suspended Solids PPM	*Particulate Data			Calcium	Sodium	Silica	Potassium	Magnesium	Alkalinity	Chlorides	Sulfates
					Size (Range)	Hardness (Ave. & Range)	Shape (General)								
46	2-23-65	RV		36											
47	2-23-65	RV		37											
48	2-23-65	RV		49											
49	2-23-65	RV		31											
50	2-23-65	RV		50											
51	2-23-65	RV		55											
52	2-23-65	RV		55											
53	2-23-65	RV		50											
54	2-23-65	RV		36											

*Supplement with occasional Micro Photograph

Comments:

California Pump
Wear Test Program

FIGURE 10-12

FIELD WATER ANALYSIS DATA SHEET

Sheet 1 of 2

Season WINTER Test Type 500

Date	Time of Day	By (Initial)	Sample No.	Sample Size	Where Taken	Turbidity	PH	Specific Conductance MICROMHOS @ 25°C	Dissolved Oxygen	Carbon Dioxide	Date Sent to Lab	Remarks: WATER LEVEL FEET	Remarks
12-16-64	1315	JW	1				7.3	719	8.1			—	
12-17-64	0115	JW	2				7.4	922	5.9			3.1	
12-17-64	1010	JW	3				7.5	797	* 11.6			3.5	
12-17-64	1500	JW	4				7.5	717	8.4			6.0	
12-17-64	1900	JW	5				7.6	722	8.0			5.8	
12-19-64	0915	SG	6				7.6	772	8.2			5.1	
12-19-64	1305	SG	7				7.6	776	8.0			5.4	
12-20-64	0015	SG	8				7.5	753	* 11.8+			5.3	EXTREME AIR BUBBLES
12-21-64	1405	SG	9			30	7.9	736	* 9.5			5.2	
12-22-64	1320	[illegible]	10				7.5	680	* 10.5			5.8	
12-23-64	1300	SG	11				7.7	630	* 10.2+			7.0	
12-24-64	0835	SG	12				7.7	650	* 9.9+			6.0	
12-25-64	2200	SG	13				7.5	730	7.3			7.6	
12-26-64	1240	SG	14				7.4	206	6.8			8.9	DIRTY WATER SHUT DOWN [illegible]

COMMENTS: * AIR BUBBLES IN HEADER — LEAK THRU SHAFT SEAL IN B-J PUMP

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Los Angeles, California
W-1

California Pump
Wear Test Program

FIGURE 10-13

Sheet 2 of 2

FIELD WATER ANALYSIS DATA SHEET

Season WINTER Test Type 500

Date	Time of Day	By (Initial)	Sample No.	Sample Size	Where Taken	Turbidity	PH	Specific Conductance (MICROMHOS @ 25°C)	Dissolved Oxygen	Carbon Dioxide	Date Sent to Lab	Remarks: WATER LEVEL FEET	
12-29-64	1525		15 III			90	7.3	153	6.4			8.9	
12-29-64	1035		15 III						8.0			8.9	
12-30-64	1035	GC	16			92	7.3	370	8.0			6.5	
12-31-64	0935	PD	17			103	7.2	243	8.2			7.0	
1-1-65	1050	GC	18			75	7.4	329	7.8			6.3	
1-2-65	1315	GC	19			103	7.4	269	8.6			6.0	
1-3-65	1410	GC	20			30	7.4	213	8.8			6.6	
1-4-65	1035	GC	21			102	7.4	337	7.6			7.1	
1-5-65	1035	PD	22			31	7.2	197	*11.0			7.2	
1-6-65	1315	JW	23			37	7.3	337	9.4			6.9	
1-7-65	1325	KV	24			30	7.3	415	8.7			7.3	
1-8-65	0915	JW	25			63	7.4	324	9.0			6.6	

COMMENTS: * AIR BUBBLES IN LINES

III NOTICE TWO SAMPLES MARKED: 15 ?

- COPY -

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Los Angeles, California

W-1

California Pump
Wear Test Program

Sheet 1 of ____

Season WINTER Test Type 1000

FIGURE 10-14

FIELD WATER ANALYSIS DATA SHEET

Date YEAR 1965	Time of Day	By (Initial)	Sample No.	Sample Size	Where Taken	Turbidity	PH	Specific Conductance MICROMHOS [illegible]	Dissolved Oxygen	Carbon Dioxide	Date Sent to Lab	Remarks WATER LEVEL FEET	
1-9	1245	DL	26			60		404	8.5			7.6	
1-10	1330	[illegible]	27			60		[illegible]	9.6			7.8	
1-11	1350	DL	28			60		172	9.1			8.2	
1-12	1300	GC	29			60		[illegible]	9.0			8.2	
1-13	1100	WF	30			60		215	9.1			6.2	
1-14	1300	[illegible]	31			60		231	9.1			7.3	
1-15	1535	GC	32			60		206	9.2			8.4	
1-16	1200	GC	33			60		264	8.9			5.4	
1-17	0400	GC	34			60		356	8.6			4.7	
1-18	1130	RV	35			40		370	9.0			5.6	
1-20	[illegible]	GC	36					394	[illegible]			5.3	
1-21	[illegible]	WF	37					457	[illegible]			[illegible]	
1-22	[illegible]	DL	38					[illegible]	[illegible]			6.0	
1-23	[illegible]	[illegible]	39					[illegible]	[illegible]			[illegible]	
1-24	[illegible]	[illegible]	[illegible]					[illegible]	9.2			5.2	

COMMENTS:

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Los Angeles, California

W-1

California Pump
Wear Test Program

Sheet 2 of ____

Season WINTER Test Type [illegible]

FIGURE 10-15

FIELD WATER ANALYSIS DATA SHEET

Date [illegible]	Time of Day	By (Initial)	Sample No.	Sample Size	Where Taken	Turbidity	PH	Specific Conductance MICROMHOS @ 25°C.	Dissolved Oxygen	Carbon Dioxide	Date Sent to Lab	Remarks: WATER LEVEL FEET	Remarks
1-25	[illegible]	GC	41					479	9.9			4.0	
1-29	1555	JW	42				7.3	234	10.1			6.9	
1-30	2000		43				7.5	345	9.3			5.9	
1-31	2200		44				7.6	427	9.3			—	
2-1	1130		45				7.6	385	9.7			4.7	
2-2	1540		46				7.6	377	9.0			5.3	
2-3	0800		47				7.6	341	9.7			5.1	
2-5	1355		48				7.4	373	9.7			5.0	
2-6	1020		49				7.4	383	9.9			6.9	
2-7	1625		50				7.6	544	9.4			4.6	
2-8	1530		51				7.5	477	9.4			5.3	
2-9	1515		52				7.6	274	9.3			6.2	
2-10	1625		53				7.3	352	10.4			5.5	
2-11	1535		54				7.4	234	10.8			6.6	

COMMENTS:

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California Pump
Wear Test Program

Sheet 1 of 6

FIGURE 10-16

STATIONARY TESTER DATA SHEET

Season WINTER Test Type 1000

Date	Time of Day	By (Initial)	Supply Pump Time Hours	Water Temp. °F	Pump Supply Press psig	Inlet Header Press psig	Outlet Header Press. psig	Diff. Press. psi	Specimen S/N, Tester No. & Flow Rate - GPM							
									-152	-252	-352	-452	-552	-652	-752	-952
									[illegible]	420 SS	304 SS	316 SS	17-4 PH	STELLITE	CHROME	EPOXY
YEAR 1965									1	2	3	4	5	6	7	8
1-8			473.0	55	550	435	0	435	4.7	6.0	6.0	6.0	5.8	5.9	5.8	5.9
1-9			497.7	51	540	432	2	430	4.0	5.3	5.8	5.9	5.8	5.8	5.2	5.6
1-10			522.4	49	550	430 / 434	2	429 / 432	4.4	6.0	6.2	6.0	5.9	5.8	5.9	5.8
1-11			545.8	49	540	437 / 434	2	435 / 432	4.5	6.0	6.0	5.9	5.9	5.8	5.8	6.0
1-12			567.9	50	540	438 / 435	3	435 / 432	4.5	6.0	6.0	4.9	5.8	5.5	5.5	6.6
1-13			591.9	48	540	405 / 434	2	403 / 432	4.5	6.0	6.0	5.0	6.0	5.5	5.8	7.5
1-14			618.0	48	540	439 / 435	3	435 / 432	4.9	6.1	6.0	4.8	5.9	5.9	5.8	8.6
1-15			644.6	49	535	436 / 435	3	433 / 432	5.0	6.2	6.1	4.5	6.0	5.8	5.6	9.0
1-16			664.9	48	535	431 / 434	2	429 / 432	4.6	6.3	6.4	4.6	5.3	5.9	5.6	9.4
1-17			680.9	48	535	425 / 434	2	423 / 432	4.7	6.3	6.1	5.0	5.7	5.5	5.5	9.7
1-18			712.4	48	530	436	2	434	4.5	6.5	6.3	5.5	6.0	5.5	5.5	11.5
1-19	1507		716.0	SHUT DOWN			SYSTEM		FOR	WEIGHING &	PHOTOS					

COMMENTS: WATER SAMPLES FOR THIS PERIOD #25 – 35

Specimen Weight - [redacted] grams

		1	2	3	4	5	6	7	8
Top Half M-5	Before	77.4653	69.1846	70.4130	70.5894	69.6336	69.9647	69.2894	68.3619
	After	77.4236	69.1647	70.3793	70.5549	69.6100	69.9616	69.2883	68.2877
	Diff.	.0422	.0199	.0337	.0345	.0236	.0031	.0011	.0742
Bottom Half M-6	Before	73.1115	65.0730	65.8023	66.7391	65.5614	66.0735	66.1638	64.3694
	After	73.0593	65.0549	65.7766	66.7126	65.5382	66.0691	66.1644	64.3080
	Diff.	.0522	.0181	.0257	.0265	.0232	.0044	.0044	.0614

Los Angeles, California

FIGURE 10-17

CALIFORNIA PUMP WEAR TEST PROGRAM
ROTATING TESTER DATA SHEET

Sheet 1 of ____

Season Winter Test Type 1000

bied W

Date 1965 Mo./Day/Year	Time of Day 1336	By (Initial) Start	Time Hour Meter Reading: Supply Pump	Time Hour Meter Reading: Tester A	Time Hour Meter Reading: Tester B	Water Temp. oF	Pump Supply Press. psig	Inlet Header Press. psig	Outlet Header Press. psig	Diff. Press. psi	Flow Rate: Tester A GPM	Flow Rate: Tester B GPM
2/18	1336	JW	—	210.7	210.5	53	560	474	39	435	160	245
19	1332	DL		234.7	234.5	53	560	472	38	434	165	249
20	1805	SG		240.6	263.0	55	558	470	38	432	169	250
21	1615	SG		failed	285.2	56	558	475	38	437	171	250
22	2145	SG			314.7	56	552	471	38	433	178	250
23	1420	WF		240.6								
	1430	WF	—			Shut	down					
26	1500	JW	.7		331.4	Start						
	1525	JW	1.1		331.8	57	550	475	38	437	180	255
27	0445	GC	1.5	345.1	345.1	56	550	472	38	434	182	258
28	1035	GC	1.1	375.0	375.0	56	545	470	37	433	183	259
3-1	1450	GC	1.7	403.3	403.3	56	543	468	38	430	186	259

COMMENTS:

Specimen Weights - Milligrams

Tester	A - Front		A - Rear		B - Front		B - Rear	
Location	Rotating	Stnry.	Rotating	Stnry.	Rotating	Stnry.	Rotating	Stnry.
Serial No.								
Before								
After								
Diff.								
Installed Clearance								
Removed Clearance								

FIGURE 10-18

Sheet 2 of ____

CALIFORNIA PUMP WEAR TEST PROGRAM
ROTATING TESTER DATA SHEET

Season Winter Test Type 1000

Date Mo./Day/Year	Time of Day	By (Initial)	Time Hour Meter Reading: Supply Pump	Time Hour Meter Reading: Tester A	Time Hour Meter Reading: Tester B	Water Temp. oF	Pump Supply Press. psig	Inlet Header Press. psig	Outlet Header Press. psig	Diff. Press. psi	Flow Rate: Tester A GPM	Flow Rate: Tester B GPM
32.65	1440	WF	96.5	427.1	427.1	56	535	37	450	423	190	260
3/3			111.5		442.2	Pump failed						

COMMENTS:

Specimen Weights - Milligrams

Tester	A - Front		A - Rear		B - Front		B - Rear		
Location	Rotating	Stnry.	Rotating	Stnry.	Rotating	Stnry.	Rotating	Stnry.	
Serial No.	251	152	151	151	551	251	451	551	
Before	1173.29	1141.55	1312.77	1137.25	1175.63	1057.49	1197.91	1066.98	?
After	1170.10	1135.15	1301.22	1130.28	1171.43	1055.08	1193.24	1062.56	
Diff.	3.19	6.40	11.55	6.97	4.20	2.41	4.67	4.42	?
Installed Clearance									

DANIEL, MANN, JOHNSON, & MENDENHALL

P-1

Los Angeles, California

California Pump
Wear Test Program

FIGURE 10-19

PHOTOGRAPHIC LOG

Sheet____ of ______

Season__________ Test Type______

Date	Photo No.	Film No.	Speci-men S/N	Description of Photo #	Comment*

Include magnification ratio if applicable.

* Film, Lighting, & Exposure Data as required.

CHAPTER 11

State of California
Department of Water Resources
Sacramento, California

California State Water Plan
Tehachapi Pumping Plant

EXPERIENCE REPORT

on

T-1 Steel

January 5, 1964

prepared by

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Baden/Switzerland

SUMMARY

This report covers the experience of Motor-Columbus with the use of T-1 steel for penstocks and pressure shafts of hydro-power stations. It describes the special design conditions, material tests, precautions for shop and field construction, welding, controls, and testing methods applied.

The report was compiled on request of the Tehachapi Crossing Consulting Board.

1. General

During its nearly 70 years of activity in hydro-power, Motor-Columous has designed and built many penstocks, underground pressure shafts, lined tunnels and manifold pipes. To recall the most important installations of the last 10 years, the power-plants of Rothenbrunnen, Safien, Sils, Barenburg, Ferrera, Motec, Vissoie, and Huinco should be mentioned, comprising 45,000 feet total length of lined tunnels, pressure shafts, and penstocks with design pressures up to 2,000 psi. Diameters vary from 6 to 19 feet, wall thicknesses from 1/2 to 2 inches, and the total weight of steel plate reaches 18,000 short tons.

Various steel qualities were used as listed below, mainly European make, but also USS T-1 steel.

Mechanical properties of some steel plate qualities (minimum values):

Trade Name	Country	Yield Strength	Tensile Strength	Elongation	Notch Test [1]	
		1000 psi	1000 psi	%	kgm/cm2	
					20°C	0°C
Aldur 35	Austria	35.6	50.0	28	8	6
Aldur 58	Austria	57.0	82.7	24	6	5
Feralsim 41	Belgium	37.0	58.3	25	10	8
Feralsim 58	Belgium	57.0	82.7	22	8	6
Soudotenax 41	Belgium	37.0	58.3	25	9	-
Soudotenax 56	Belgium	57.0	80.0	20	7	-
Union BH 36	Germany	51.2	74.0	24		7.5 [2]
Union 54	Germany	64.0	82.8	20	6	
USS T-1 [3]	USA	88.2	106.0-124.2	18		5 [2]
USS T-1	USA	100.0	114.0-140.0	15		5 [2]

1) Swiss Standard Test
2) At -10°C
3) Before 1959

T-1 steel was used for the manifold of the Sils power station and for parts of the Sils and Barenburg pressure shafts. Characteristic dimensions of these installations are as follows:

Sils manifold:	Main pipe diameter	11 - 5 1/2 ft.
	Main pipe length	300 ft.
	Main pipe wall thickness	1 5/16"-7/8"
	Branch pipes, number	4
	diameter	5 1/2 ft.
	wall thickness	7/8"
	length, each	30 ft.
	Design pressure	680 psi
Sils pressure shaft:	Diameter	14 - 11 ft.
	Wall thickness	3/4" - 1 5/16"
	Design pressure	670 psi
Barenburg pressure shaft:	Diameter	13 - 11 ft.
	Wall thickness	3/4" - 1 1/2"
	Design pressure	600 psi
	Total weight T-1 steel	900 short tons

The following paragraphs are an abstract of the general specifications for pressure conduits and of the specific requirements stipulated for T-1 steel.

2. Calculated stresses and safety factors

For straight lengths of pipes the safety factor is usually 2.0, based on the design pressure and the yield point of the material. The design pressure equals the maximum static pressure plus water-hammer.

For pipes consisting of other configurations than straight lengths of cylindrical shells, a higher safety factor is used. This applies to bends, manholes, branches, etc. For underground pressure tunnel linings, the wall thickness is diminished by a factor which defines how much of the internal pressure is taken over by the rock. On the other hand, the possibility of outside pressure has to be considered for underground linings.

An extra increase of the safety factor because of T-1 steel was not considered necessary.

3. Material

3.1 Choice of Material

The choice of material is left open to the contractor within the following limitations:

> "Only high-grade and age resisting open hearth steel plates with sufficient cohesive strength shall be employed, free from laminations and other rolling mill faults. The tenderer is free to offer the material which appears to him to be suitable and economical for the duty involved, whereby he shall prove that he has sufficient practical experience of working and welding this material."

According to these general terms, the contractor will offer different steel qualities for the same type of conduit. After a detailed comparison on technical quality and economy, the best solution is chosen.

3.2 Material Controls

Tests and controls are applied on the steel plates in the rolling mill, on the electrodes, on the welds, and on the finished parts in the contractor's shop as well as in the field.

In the rolling mill:

a. For each plate

1 elongation test, in the delivered condition, across the rolling direction

3 notch toughness tests, in the delivered condition, at $-10^{o}C$, along the rolling direction

3 notch toughness tests in an aged condition, at $+20^{o}C$, along the rolling direction

1 elongation test, across the rolling direction, at the foot

surface check, dimensions check, supersonic tests with respect to lamination and rolling mill defects

The plates shall be stamped (melt no., plate no., quality designation, examiner's stamp).

b. Per melt

1 chemical analysis concerning C, Si, Mn, S, P, possibly Cr, Cu, Ni.

Before starting work in the workshops, material test certificates from the rolling mill must be submitted.

At the contractor's works:

a. Plate tests

Snatch tests shall be carried out at the contractor's works, at least one test for each quality of material (thickness class and melt).

These tests include:

1 chemical analysis

1 elongation test along the rolling direction, in the delivered condition

1 elongation test along the rolling direction, stress-free annealed, for plates whose tensile strength is greater than 50 kg/cm2

3 notch toughness tests, in the delivered condition, at -10°C, along the rolling direction

3 notch toughness tests in an aged condition, at +20°C, along the rolling direction

3 notch toughness tests, in the delivered condition, at +20°C, across the rolling direction

1 bending test, along the rolling direction

1 diagram showing the notch sensitivity in function of the temperature, for material in the delivered condition, along the rolling direction

1 diagram showing the notch sensitivity in function of the temperature, for material in an aged condition, along the rolling direction.

Before being used, each sheet shall be subjected to a brinell hardness test in order to control the quality of the material.

b. Mechanical welding seam tests

Extensions of longitudinal seams shall be carried out for testing purposes in the contractor's works, at least one for each quality of material (thickness class and melt).

The tests include:

2 elongation tests with parallel test pieces

1 elongation test with shaped test piece that causes a fracture in the welding metal

2 bending tests

3 notch toughness tests at 20°C

1 diagram showing the notch sensitivity in function of the temperature in the welding metal

1 diagram showing the notch sensitivity in function of the temperature in the transition zone.

c. X-ray and supersonic tests

Usually 20% of the shop and field welds are subject to ultrasonic tests. X-ray photographs are made at random, at least of 2% of all welding seams, plus of all those welds where the ultrasonic test has shown some suspicions of fault.

For the T-1 manifold, the ultrasonic tests were extended to 100% of all shop and field welds. X-ray tests were made as above.

4. Design and Manufacture

In principle, the design of the structure has to follow the special requirements of the used material with respect to manufacturing, bending, welding, etc. Such rules must be followed especially for quenched and tempered

steel which cannot be machined with usual methods and which require special care with respect to any heat treatment applied.

For high stress steels like T-1 special instructions should be stipulated for the design, manufacturing, and welding, and it is important that these instructions are followed exactly. These instructions should comprise the following points:

a. Quenched and tempered steels are delivered from the rolling mill with a specific heat treatment. They are hardened and tempered in a special treatment to reach the guaranteed stress values.

b. For the design it is important that the structure consists only of geometrical plane, cylindrical and tapered shells. Two-dimensional curvatures and spherical parts must be avoided if a heating of the steel up to 1100°C would be necessary to achieve such shape.

c. Bending of the plates has to be made under cold conditions, either on a hydraulic press or a rolling mill. The deformation of the plate should be restricted to maximum 6% elongation.

d. Ultrasonic tests should be applied on the plates in the neighborhood of all welding seams to assure that no laminations or other rolling faults are in the parts near the welding seams. Laminations extend into the weld during the welding and cause cracks parallel to the plate which are often not to be seen in X-ray photographs.

e. After torch cutting the welding edges must be machined or ground (in the field) to eliminate that zone which has been heated too much by the torch cutting.

f. Any welding should be made only with the special electrodes which have been selected for the material.

g. The plate ends have to be preheated before welding and have to be kept on the specified temperatures during the welding. After welding, the welding seams should be cooled gradually in calm air.

h. The welding seams should be ground inside and outside, plane and smooth to the plate surface. This will allow to find out burnt-in spots or notches.

i. If a new material is applied the first time, model tests should be made to prove that the manufacturing and design methods and instructions cover all requirements. Such model tests on Y-pieces of manifolds consist of hydraulic tests to measure the hydraulic friction losses, and of stress tests. The stress tests are necessary to control the calculation by strain gauge measurements. If a new material is applied, the stress tests should be carried on to the rupture of the element, to verify the behavior of the material through the full range of stresses up to the tensile stress. It controls also the quality of the weldings. The rupture should not start in a welding seam.

5. Shop and Field Work with T-1 Steel

5.1 Bending, Rolling and Cutting

No problems were experienced. All bending and rolling work was done under cold conditions.

5.2 Electrodes

After extensive tests with different types, "Ultratherm" electrodes, made by Secheron, Geneva, were selected. "Ultratherm" electrodes are of the basic type, with thick coating, low carbon content. One-half per cent Molybdenum and 3.5% Nickel content are used to achieve strength requirements. This electrode was developed especially for T-1 steel welding. It was also used for welding of T-1 to other type steels.

The diameter of the electrodes was limited to maximum 5 mm, to achieve that the weld is built up in small layers.

5.3 Welding Procedure

Because of the cold climate in Switzerland, special care was necessary with respect to notch toughness. Extensive tests had shown that a notch toughness as high as 5 kgm/cm2 at -20°C in the weld could be reached with hand-welding. With automatic welding the notch toughness was only 4 kgm/cm2 at 0°C. The excellent result of the hand-welding is caused by the small electrode diameter. Each layer gives a proper heat treatment to the foregoing layer. For that reason it was decided to do all welding, in the shop and in the field, by hand. Metallurgical investigations proved that the weld itself and the border zones of the plate consist of fine-grain Martensit structure of low carbon content.

Before use, the electrodes were dried at 300°C for 2 - 4 hours. The oven was installed close to the welders to ensure that only dry electrodes were used.

The plates were preheated to 130 - 150°C. During the welding the plate temperature was checked continuously to ensure that the temperature never dropped below 80 - 100°C. This was done by means of temperature indicating chalks.

Cooling down of the finished welds was done without special heating, but precautions were necessary to avoid draught.

Only welders with welding examination were employed. The welders had to mark all their welds. Thus, the 100% ultrasonic welding control also provides a continuous check of the welding personnel.

Field welding requires protection against wind and draught, rain, and splashing water. Roofs and tent-cloth were used.

5.4 Heat Treatment

Manifold pipes, made of high-stress steels with yield point around 50, 000 psi, are usually stress released after welding. This is necessary because of the relatively great wall thickness of such structure.

With T-1 steel the wall thickness for the same structure is only approximately one-half due to the yield point around 100, 000 psi. This reduction in wall thickness makes the structure much more elastic, and the amount of welded-on material is reduced to about 1/4 per weld. The internal stresses introduced by welding are, therefore, much less than with material of lower yield point, and a stress release heat treatment can be avoided.

Any heat treatment is considered dangerous because of the possible negative effect on the quenched and tempered T-1 steel. It was decided that no stress releasing should be made, neither on the shop-fabricated parts nor on field welds. This decision was important. Experience with other T-1 steel manifolds showed that stress release heating may indeed lead to failure.

5. 5 Pressure Test

The Sils manifold was tested as a whole after complete erection in the field. The test pressure was 1. 5 times the design pressure. During the test the stresses in the manifold were measured with strain gauges. The pressure was gradually raised to test pressure, then reduced to design pressure and all welding seams hammered with a copper hammer, two pounds in weight. After that, test pressure was applied again. No leakage or fissure was found. Strain gauge readings confirmed that the actual stresses were as calculated.

For the pressure shafts of Sils and Barenburg, the tightness was checked by filling tests. Again, no leakage was found.

6. Failure of T-1 Manifolds

While the Sils manifold was a full success, two other T-1 steel manifolds in Switzerland failed. Motor-Columbus was not concerned with these cases.

Failures occurred (a) with parts where two-dimensional curvatures were made by heating the plate (approximately 1100°C) and (b) in the neighborhood of field welds. The failure itself consisted of small fissures and cracks at these points. We are not informed about the field welding procedure and electrode types, but the field welds have been stress released.

The Swiss Federal Institute of Technology (ETH), Zurich, investigated the failing manifolds thoroughly. The following points are an abstract of their findings and recommendations, which also confirm that the welding and manufacturing instructions applied in case of the Sils manifold were correct:

a. Preheating of the plates for welding is recommended with temperatures of 80 - 100°C.

b. For multiple-layer welds the weld itself should have the same temperature as the plate (70 to 100°C) when the next layer is applied.

c. Proper drying process of the electrodes in an oven immediately before use.

d. No stress releasing (600 - 650°C) should be made.

Under strict obedience to these rules, T-1 steel can be safely welded, says the ETH-report.

7. Conclusions

Motor-Columbus has applied T-1 steel for the manifold and for parts of the pressure shaft at Sils power station, and for parts of the pressure shaft of the Barenburg power station. Design and manufacture were made under strict observance of instructions especially developed for the use of T-1 steel through extensive metallurgical, physical, welding, and stress model tests. The use of T-1 steel was successful and no failure occurred.

There were failures with two other manifolds of T-1 steel in Switzerland. Investigation of these cases by the Swiss Federal Institute of Technology (ETH) resulted in welding and fabrication instructions similar to those applied with the Sils manifold. The failing manifolds were repaired successfully following these instructions.

This leads to the conclusion that T-1 steel is suitable for high-pressure hydraulic penstocks and shaft linings. It does, however, require special methods for design, manufacture and welding. Controls of all fabrication procedure must ensure that the instructions are followed strictly.

MOTOR-COLUMBUS
Electrical Management Company Ltd.

January 5, 1964

CHAPTER 12

REFERENCE MATERIAL FROM MODEL TEST FIRMS

Material prepared for this spot falls within the category of information considered proprietary by Daniel, Mann, Johnson, and Mendenhall and has been deleted at the request of that firm.